Renewable and Sustainable Energy Reviews 15 (2011) 4013–4029

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Transcritical carbon dioxide heat pump systems: A review

Brian T. Austin, K. Sumathy*

Department of Mechanical Engineering, North Dakota State University, Fargo, ND 58103, USA

ARTICLE INFO

Article history:
Received 28 December 2010
Accepted 5 July 2011
Available online 21 August 2011

Keywords:
Transcritical cycle
Heat pump
CO_2
Natural refrigerant

ABSTRACT

Carbon dioxide is a safe, economic and environmentally sustainable refrigerant which can be used in heat pump and refrigeration systems. Research into the performance and benefits of a transcritical heat pump cycle using carbon dioxide began in the early 1990s. Theoretical and experimental research, as well as commercial system development, has improved transcritical system performance to a level similar to that of conventional heat pump systems. This paper presents an overview of transcritical carbon dioxide heat pump systems. The paper begins with a summary of carbon dioxide's use as a refrigerant and the distinctions of the transcritical cycle, followed by a numerical analysis of transcritical cycle performance. The study will then present a review of research on transcritical carbon dioxide heat pump systems, which covers system components, configurations and modifications and how these factors affect overall system performance.

Published by Elsevier Ltd.

Contents

* Corresponding author at: North Dakota State University, Dept 2490, PO Box 6050, Fargo, ND 58108, USA. Tel.: +1 701 231 7139; fax: +1 701 231 8913.
E-mail address: sumathy.krishnan@ndsu.edu (K. Sumathy).

1364-0321/$ – see front matter. Published by Elsevier Ltd.
doi:10.1016/j.rser.2011.07.021

Nomenclature

A	heat transfer area, m^2
COP	coefficient of performance
c_p	specific heat, kJ/kg K
d	diameter
G	mass flux, kg/m^2
h	specific enthalpy, kJ/kg
L	length, m
$\dot{m}$	mass flow rate, kg/s
N	compressor speed, Hz
Nu	Nusselt number
P	pressure, Pa
Pr	Prandl number
Q	heat transfer rate, W
q_w	heat flux through the tube wall, W/m^2
r	compressor pressure ratio
Re	Reynolds number
T	temperature, °C
ΔT_{ap}	approach temperature difference, °C
U	overall heat transfer coefficient, W/m^2 K
V_s	swept volume of compressor, m^3
W_{comp}	compressor work, W

Greek letters

f	friction factor
f_c	friction factor at constant thermophysical properties
η	efficiency
μ	dynamic viscosity, N/m^2 s
ξ	minor loss coefficient
ρ	density, kg/m^3

Subscripts

b	bulk temperature
comp	compressor
crit	critical
ev	evaporator
exp	expansion device
gc	gas cooler
i	inlet; inner
is	isentropic
o	outlet
pc	pseudo critical
v	volumetric
w	water; wall temperature

1. Introduction

Climate change is a major worldwide concern with potentially dramatic impacts on developing and industrialized countries alike. The effort to mitigate climate change centers on the reduction of greenhouse gas emissions. Carbon dioxide is the greenhouse gas which receives the most attention due to sheer volume of emissions, much of it from power generation. However, other atmospheric pollutants such as methane and nitrous oxide have a far greater impact on climate change on a per mass basis. The global warming potential (GWP) is a relative measure of the heat trapping effect of a gas in comparison to an equal mass of carbon dioxide over a given quantity of time in the atmosphere [1]. Methane and nitrous oxide have GWP values of 23 and 296 respectively, both based on a 100 year time span [2].

The goal of reducing greenhouse gas emissions has become a major driver of technology. Transportation, energy production, agriculture and manufacturing sectors have all sought technological changes in order to reduce emissions. The heating, ventilation, air conditioning and refrigeration (HVAC&R) industry is also developing systems with lower impacts on climate change.

Many of the refrigerants used in HVAC&R systems are potent greenhouse gases. R134a, for instance, has a GWP of 1300 over a 100 year time span [2]. When securely contained in a properly operating system refrigerants do not impact climate change; however, system leaks and improper recovery of refrigerants during repairs or at end of life result in these harmful gases entering the atmosphere. Some climatologists have called for a complete worldwide phase-out of refrigerants with high GWP similar to the phase-out of ozone depleting substances enacted under the Montreal Protocol in 1987. Already, the European Union has approved the scheduled phase-out of mobile air conditioning systems using refrigerants with GWP greater than 150. This directive was ratified in 2007 and went into effect beginning in 2008 [3,4].

One potential replacement refrigerant is carbon dioxide, a natural refrigerant which has negligible impact on climate change. Carbon dioxide used in HVAC&R systems has a zero net impact on climate change because it has been recovered from other industrial processes [5]. Furthermore, carbon dioxide (CO_2) is not toxic, flammable or corrosive, and it has no impact on the ozone layer. It is inexpensive and readily available. CO_2's performance as a refrigerant in heat pump systems is also competitive with refrigerants currently in use [5–7].

2. Unique properties of carbon dioxide

Two factors require special attention when using CO_2 as a refrigerant for heat pump systems. One is the low critical temperature. The other is the high working pressure required to use CO_2 under typical heat pump conditions.

Carbon dioxide becomes a super critical fluid at 31.1 °C at 73.7 bar. In a conventional (subcritical) heat pump cycle, low critical temperature (T_{crit}) is a disadvantage because it limits the operating temperature range; heat cannot be delivered at temperatures greater than the critical temperature. Further, at temperatures less than but near T_{crit}, the enthalpy of vaporization is reduced. This leads to a reduction in heating capacity and poor performance of the system [6]. Thus a conventional heat pump should avoid operating at a heat rejection temperature near T_{crit}. In a transcritical heat pump, heat rejection pressures are greater than the supercritical pressure and heat delivery temperatures are no longer limited by T_{crit}. CO_2's low critical temperature provides the opportunity to operate in a transcritical manner.

High working pressure is the other notable distinction of CO_2 heat pumps. Both subcritical and transcritical heat pump systems using CO_2 operate at pressures greater than with most other refrigerants. Subcritical CO_2 heat pumps may function at pressures as high as 60–70 bar, while transcritical systems may have pressures from 80 to 110 bar or more. For comparison, R134a has a saturation pressure of 13.18 bar at 50 °C [8].

High pressure presents design challenges in terms of component robustness and compressor capability; however, today's manufacturing capabilities allow production of components which can meet these demands. In addition, high pressure presents some benefits: CO_2 has a relatively high vapor density and correspondingly a high volumetric heating capacity. This allows a smaller volume of CO_2 to be cycled to achieve the same heating demand which allows for smaller components and a more compact system [5,6].

3. A brief history of CO_2's use as a refrigerant

Heat pump systems are closely related to refrigeration systems, but vapor compression cycles were developed for refrigeration long

before the concept was applied to heating. Likewise, refrigerant fluids were developed for refrigeration rather than heating. A detailed history of CO_2's role in refrigeration development was given by Pearson [9].

Carbon dioxide was among the first refrigerants used in commercially viable vapor compression refrigeration systems. Other early refrigerants included ether, ammonia, sulfur dioxide and methyl chloride. CO_2 was first used in a vapor compression system to produce ice by Thaddeus Lowe in 1866. The high working pressures of CO_2 were a hindrance to implementation, allowing ammonia and sulfur dioxide systems to become established first. Ammonia systems were generally efficient, but they were quite large and ammonia's toxic nature presented a safety hazard. In comparison, CO_2 required more fuel and more robust components, but the systems were much smaller and CO_2 leaks did not pose a health risk.

For its improved safety and compactness ships began to use CO_2 for refrigeration in the 1880s and 90s, while on land ammonia was dominant. A further reason for the divergence of land and sea systems respectively to ammonia and CO_2 was condensing temperature and means of condensing. Due to CO_2's low critical temperature, significant loss of capacity and efficiency occur as condenser temperature increases. Ammonia can operate under a wider range of condenser temperatures. Early systems mainly used river or sea water as the heat sink for condensing. This provided the low temperatures required for CO_2. Later, evaporative coolers were utilized when a body of water was unavailable. Evaporative coolers generally produce higher condensing temperatures which favored ammonia over CO_2.

The land and sea trend continued into the early 1900s when ammonia's improved safety record and better manufacturing caused acceptance of ammonia refrigeration on ships. By the 1930s ammonia plants were preferred even at sea and the use of CO_2 for refrigeration further decreased. The final demise of CO_2 in refrigeration was caused by synthetic refrigerants. R12 and R11 were first introduced for commercial use in 1931 and 1932 [10]. They were non-toxic, non-flammable and operated efficiently over a range of temperatures. Synthetic refrigerants began to displace CO_2 and came to dominate non-industrial systems by the 1950s and 60s.

Interest in CO_2 was renewed in the early 1990s in part due to the phase-out of ozone depleting refrigerants. Norwegian professor Gustav Lorentzen has received much of the credit for the new attention given to CO_2, however, there were others studying CO_2 at the same time. Lorentzen published a patent application for a trans-critical CO_2 automotive air conditioning system in 1990 [11]. Lorentzen's transcritical cycle eliminates the problem of capacity and efficiency loss that subcritical systems have when operating with heat rejection temperatures near the critical point. Technological and manufacturing improvements make it possible now to achieve the high pressures required for transcritical operation. One of the first transcritical CO_2 systems was a prototype automotive air conditioning system built and tested by Lorentzen and Pettersen [12]. The system was further reported by Pettersen [13]. Performance was similar to that of an R12 system and encouraged the further development of the transcritical CO_2 system.

Research into CO_2 refrigeration, air-conditioning and heat pump systems continues, but some CO_2 systems have already been successfully commercialized. For nearly a decade, transcritical CO_2 heat pump water heaters have been commercially available in Japan. Introduced in 2001, over 1 million of the EcoCute water heaters had been sold by 2007 [14] and sales topped 2 million in October of 2009 [15]. Vending machines using a transcritical CO_2 refrigeration cycle are becoming more common in Japan and throughout Europe. In December of 2009 The CocaCola Company announced it would phase out fluorinated refrigerants in all new vending machines by 2015, switching primarily to CO_2 systems [16]. In addition, CO_2 commonly serves as the low temperature refrigerant in cascade type industrial refrigeration systems, and CO_2 is increasingly used as a secondary fluid in food display applications where harmful refrigerants must be kept separate for safety reasons [10].

4. Comparison of conventional and transcritical heat pump systems

Unlike combustion-based and electrical resistance heating systems, a heat pump does not generate heat. All heat pumps, whether conventional (subcritical) or transcritical provide heat by transferring it from one zone to another. The delivered heat energy can be several times greater than the work input to the heat pump. This stands in contrast to combustion type heating systems which deliver much less useful heat than the energy contained its fuel.

The concept of a heat pump is often credited to Lord Kelvin, but he did not demonstrate the concept [17]. The first commercial heat pump installation was in the Equitable Building of Portland, Oregon in 1948 [18]. Despite their potential for energy savings, heat pumps have never been the dominant method of space heating.

In a conventional heat pump, the entire cycle occurs below the critical point of the refrigerant being used. Heat absorption occurs by evaporation of the refrigerant at low pressure, and heat rejection takes place by condensing the high pressure refrigerant. In a transcritical cycle an evaporator still serves the heat absorption function, but heat rejection is not through condensation. The refrigerant pressure is increased into the supercritical region, and heat rejection occurs by single-phase sensible cooling (gas cooling). Fig. 1 shows the distinction between a subcritical and a transcritical cycle on separate *P*–*h* diagrams. Heat rejection takes place via the gas cooler rather than a condenser.

To make effective use of the transcritical cycle, the large pressure difference and the uniqueness of the gas cooling process must be addressed. Optimization of the transcritical cycle depends on the components and various operating parameters which are different from the conventional cycle. Certain heating applications benefit from the unique characteristics of the transcritical cycle. An example is heating applications that require a very large temperature increase. Since the gas cooler rejects heat by sensible cooling, the difference between the inlet and outlet temperatures (temperature glide) is much greater than in a condensing process. Thus the transcritical cycle is more beneficial for heating applications requiring large temperature increase.

The pressure difference between the heat rejection pressure (gas cooler pressure) and the heat absorption pressure (evaporator pressure) is much greater in a transcritical CO_2 system than in a typical subcritical system. This results in large thermodynamic losses during the expansion process. However, the large pressure difference also makes it feasible to implement an expansion work recovery device into the system. Expansion work recover could partially compensate for the large the throttling losses of transcritical CO_2 heat pumps [19]. It should also be noted that the transcritical system's pressure ratio is actually lower than that of the many conventional systems. Transcritical CO_2 systems typically operate at a pressure ratio of three or four while an R134a system may operate with a pressure ratio up to eight [20]. The lower pressure ratio allows the transcritical system's compressor to operate with greater efficiency.

5. Modeling and analysis

Modeling of a transcritical carbon dioxide heat pump system is conducted based on thermodynamic analysis of the system and based on the transport characteristics of the refrigerant and

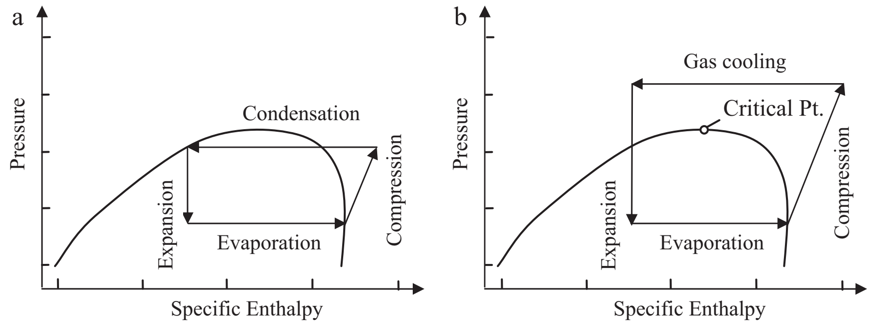

Fig. 1. *P–h* diagrams showing: (a) subcritical cycle and (b) transcritical cycle.

secondary fluids. The governing thermodynamic equations for each component of a basic transcritical heat pump are presented in Sections 5.1.1–5.1.4 while the heat transfer and pressure drop characteristics of CO_2 are presented in Sections 5.2.1–5.2.3.

Modeling of heat pump systems usually involves certain simplifying assumptions. Typical assumptions for transcritical CO_2 heat pumps include:

- The system is operating at steady state.
- Changes in kinetic and potential energy are negligible.
- Compressor operates adiabatically.
- All heat exchangers operate adiabatically.
- Heat loss in connecting piping is negligible.
- The expansion process is isenthalpic.

5.1. Thermodynamic analysis

Heat pump performance is typically discussed in terms of heating capacity and coefficient of performance (COP). Heating capacity is the amount of heat delivered by the system. It is calculated as shown in Eq. (1) where $\dot{m}$ is the refrigerant mass flow rate and $h_{gc,i}$ and $h_{gc,o}$ are the refrigerant enthalpy values at the gas cooler inlet and outlet respectively. Coefficient of performance (COP) can be defined in terms of heating or cooling. For heating, it is the ratio of heat output to compressor work. The cooling COP is the ratio of heat removed over compressor work. COP values are calculated as shown in Eqs. (2) and (3).

$$Q_{capacity} = \dot{m} \times (h_{gc,i} - h_{gc,o}) \tag{1}$$

$$\mathrm{COP}_{heating} = \frac{h_{gc,i} - h_{gc,o}}{h_{comp,o} - h_{comp,i}} \tag{2}$$

$$\mathrm{COP}_{cooling} = \frac{h_{ev,o} - h_{ev,i}}{h_{comp,o} - h_{comp,i}} \tag{3}$$

The analysis of heat pumps generally assumes steady refrigerant flow; therefore, for all components, inlet and outlet flow rates are equal.

$$\dot{m}_i = \dot{m}_o = \dot{m} \tag{4}$$

Each component of the system can be defined by an energy balance equation. For clarity, the following section will develop equations assuming both the gas cooler and evaporator are concentric-tube, counter-flow heat exchangers with CO_2 flowing in the inner tube and water used as the secondary fluid.

5.1.1. Gas cooler

The gas cooler is typically modeled as a concentric tube counter flow heat exchanger. Energy balance equations can be defined for both the refrigerant side and the water side as shown in Eqs. (5) and (6) respectively.

$$Q_{gc} = \dot{m} \times (h_{gc,i} - h_{gc,o}) \tag{5}$$

$$Q_{gc} = \dot{m}_w \times c_{p,w} \times (T_{gc,w,o} - T_{gc,w,i}) \tag{6}$$

Heat transfer in a concentric tube gas cooler is generally considered to occur only in the radial direction and heat transfer along the tubes is negligible. The temperature gradient along the tubes is much greater in a gas cooler than in a conventional condenser or evaporator. This has raised some question as to the suitability of assuming negligible longitudinal heat transfer. In a numerical study Asinari et al. [21] determined that, even in the regions with the greatest temperature gradient, the impacts of longitudinal heat flow in gas cooler tubes is negligible.

The heat transfer rate, Q_{gc}, can be defined based on the overall heat transfer coefficient and the temperature difference between the two fluids. Heat transfer rate is calculated in Eq. (7) using the logarithmic mean temperature difference method. In this equation U is the overall heat transfer coefficient and A is the heat transfer area.

$$Q_{gc} = UA \times \frac{(T_{gc,i} - T_{gc,w,o}) - (T_{gc,o} - T_{gc,w,i})}{\ln(T_{gc,i} - T_{gc,w,o})/(T_{gc,o} - T_{gc,w,i})} \tag{7}$$

5.1.2. Evaporator

Energy balance and heat transfer equations for the evaporator are similar to that of the gas cooler, as shown in Eqs. (8)–(10).

$$Q_{ev} = \dot{m} \times (h_{ev,o} - h_{ev,i}) \tag{8}$$

$$Q_{ev} = \dot{m}_w \times c_{p,w} \times (T_{ev,w,i} - T_{ev,w,o}) \tag{9}$$

$$Q_{ev} = UA \times \frac{(T_{ev,w,i} - T_{ev,o}) - (T_{ev,w,o} - T_{ev,i})}{\ln(T_{ev,w,i} - T_{ev,o})/(T_{ev,w,o} - T_{ev,i})} \tag{10}$$

5.1.3. Compressor

With the assumption that the compressor is operating adiabatically, the compressor work can be calculated as shown in Eq. (11).

$$W_{comp} = \frac{\dot{m}}{\eta_{is}} \times (h_{discharge,is} - h_{suction}) \tag{11}$$

The mass flow rate ($\dot{m}$) can be related to compressor operation as defined in Eq. (12), where V_{swept} is the swept volume of the compressor, η_v is the volumetric efficiency of the compressor, N is the compressor speed and $\rho_{suction}$ is the density of the refrigerant at the suction port.

$$\dot{m} = V_s \times \eta_v \times N \times \rho_{suction} \tag{12}$$

The volumetric and isentropic efficiencies are both functions of the compressor pressure ratio (r) and can be estimated by the correlation given in Eqs. (13) and (14) [22].

$$\eta_v = 0.9207 - 0.0756 \times (r) + 0.0018 \times (r)^2 \tag{13}$$

$$\eta_{is} = -0.26 + 0.7952 \times (r) - 0.2803 \times (r)^2 + 0.0414 \times (r)^3 - 0.0022 \times (r)^4 \tag{14}$$

5.1.4. Expansion device

The energy balance for the expansion device is straightforward since it is being modeled as isenthalpic.

$$h_{\exp,i} = h_{\exp,o} \tag{15}$$

5.2. CO_2 transport characteristics

In the subcritical and supercritical regions, the magnitude and variation of transport properties of CO_2 (viscosity and thermal conductivity) and other properties are significantly different from that of other fluids. Consequently, well established correlations which relate the heat transfer coefficient and pressure drop under different conditions are inaccurate for CO_2. In order to create sufficient numerical models of transcritical heat pump systems using CO_2, accurate correlations are needed to predict the heat transfer coefficient and pressure drop during supercritical gas cooling, single phase heating and cooling, and flow boiling processes.

5.2.1. Heat transfer correlations: supercritical cooling

Many of the properties of supercritical carbon dioxide vary widely with temperature and pressure. Property variation is greatest in a region near the pseudo critical temperature. The pseudo critical temperature (T_{pc}) is defined as the temperature at which the specific heat of CO_2 reaches a maximum. Eq. (16) can be used to determine T_{pc} of CO_2, which is proportional to the pressure condition [23].

$$T_{pc} = -31.40 + 12.15P - 0.6927P^2 + 0.03160P^3 - 0.0007521P^4 \tag{16}$$

The variation in CO_2's properties impacts the heat transfer and flow characteristics including viscosity, thermal conductivity, specific heat and density. These variations must be taken into account in heat transfer correlations of supercritical CO_2 during cooling process. It should be pointed out that correlations developed for supercritical heating do not accurately predict the heat transfer during supercritical cooling.

Early development of a supercritical heat transfer correlation for CO_2 was conducted by Krasnoshchekov et al. [24] on turbulent flow, and Baskov et al. [25] later found that the correlation over predicted their experimental results. Petrov and Popov [26] developed a new correlation by modifying an earlier Nusselt correlation of Petukhov and Popov [27].

More recently the subject of cooling supercritical CO_2 heat transfer correlation in horizontal channels and microchannels has been investigated by several researchers [28–38]. Yoon et al. [28] conducted an experimental study to determine the heat transfer coefficient for CO_2 flowing in a 7.73 mm horizontal tube. Based on the experimental results a new correlation was proposed which was a modification of the Baskov correlation [25]. A study by Oh and Son [39] compared the correlations of several recent studies and found the correlation of Yoon et al. [28] to be one of the most accurate for macro-channels.

The correlation of Yoon et al. [28], as shown in Eq. (17) has two sets of parameters that apply respectively to temperatures greater than T_{pc} and temperatures less than or equal to T_{pc}. Many correlations require properties to be evaluated at the bulk temperature and at the wall temperature. For engineering and design purposes, the bulk temperature is typically known, but the wall temperature of the heat exchanger is unknown. The correlation of Yoon et al. [28] is more applicable for engineering purposes since it uses the bulk temperature for all property evaluation.

$$\begin{gathered} \mathrm{Nu}_b \quad a\mathrm{Re}_b^b Pr_b^c \left(\frac{\rho_{pc}}{\rho_b}\right)^n \\ \text{for} \quad T_b < T_{pc} : a = 0.14, \quad b{=}0.69, \quad c{=}0.66, \quad n{=}0 \\ \text{for} \quad T_b \le T_{pc} : a{=}0.013, \quad b{=}1.0, \quad c{=}-0.05, \quad n{=}1.6 \end{gathered} \tag{17}$$

5.2.2. Pressure drop correlations: supercritical flow

The pressure drop for supercritical CO_2 can be expressed by the Darcy–Weisback equation for single-phase pressure drop as shown in Eq. (18), where f is the friction factor and ξ is the minor loss coefficient.

$$\Delta P = \frac{G_2}{2\rho}\left(f\frac{L}{d_i} + \xi\right) \tag{18}$$

The friction factor has been defined by Blasius for turbulent flow in smooth tubes as shown below:

$$f = \frac{0.316}{Re^{1/4}} \quad \text{for}\, Re \le 2 \times 10^4 \tag{19a}$$

$$f = \frac{0.184}{Re^{1/5}} \quad \text{for}\, Re \le 2 \times 10^4 \tag{19b}$$

Petrov and Popov [40] developed a correlation for the pressure drop of cooling supercritical CO_2 and defined the friction factor as in Eqs. (20a) and (20b).

$$\begin{gathered} f \quad f_{c,w}\frac{\rho_w}{\rho_b}\left(\frac{\mu_w}{\mu_b}\right)^s \\ \text{for} \quad 1.4 \times 10^4 \le \mathrm{Re}_w \le 7.9 \times 10^5 \\ \text{for} \quad 3.1 \times 10^4 \le \mathrm{Re}_b \le 8.0 \times 10^5 \end{gathered} \tag{20a}$$

$$s = 0.023\left(\frac{|q_w|}{G}\right)^{0.42} \tag{20b}$$

In this expression the subscripts w and b signify properties evaluated at the wall and bulk temperatures respectively. $f_{0,w}$ is the friction factor at constant thermophysical properties.

More recently, several authors [28,29,32,41,42] have analyzed the pressure drop of cooling supercritical CO_2 in order to compare the results to established correlations. Cheng et al. [43] reviewed the pressure drop findings of several authors and compared the results. Cheng et al. concluded that the Blasius equation for friction factor predicts the pressure drop of cooling supercritical CO_2 in both micro- and macro-channels with sufficient accuracy.

5.2.3. Heat transfer and pressure drop correlations: flow boiling

The pressure drop and heat transfer characteristics of CO_2 during flow boiling have been analyzed in many research works [44–52]. The results of the studies have typically concluded that the generalized correlations for pressure drop and heat transfer do not adequately predict the behavior of CO_2. This has led some authors to develop new predictive correlations specific to CO_2.

Mastrullo et al. [53] compared experimental data to the pressure drop and heat transfer predicted by several correlations. The correlations of Cheng et al. [54] and Jung et al. [55] were found to most accurately predict the heat transfer coefficient of CO_2. The correlation of Cheng et al. was developed specifically for CO_2 while Jung et al.'s correlation was developed as a general fluid correlation. The

generalized pressure drop correlations of Muller-Steinhagen and Heck [56] and Friedel [57] provided the closest prediction of the experimental data.

6. Performance of transcritical CO_2 heat pump systems

A heat pump is generally used to heat either air or water. Air, for example, may be heated for the purpose of space heating, while water may be heated for domestic hot water or spacing heating. The use of air or water as the heat recovery fluid affects the heat flux at the gas cooler due to the differing thermophysical properties of the fluids. Likewise, the evaporator may be absorbing heat from different sources such as ambient air, water from a lake or the ground, which will impact the heat exchange. These factors, directly or indirectly, will influence the operating parameters and hence will impact the performance of a heat pump. This review includes theoretical and experimental works conducted with both air and water as the heat source and heat sink fluids.

An air conditioning or refrigeration system is essentially the same as a heat pump except that the desired output is different and the operating temperatures are different. Because the systems are similar, research on transcritical CO_2 air conditioning and refrigeration systems can be instructive for transcritical CO_2 heat pump development as well. For this reason, studies on systems designed for cooling purposes have been included in this review. It must be taken into account that the COP and capacity are defined differently for cooling systems. Cooling capacity is the amount of heat absorbed by the evaporator, while $COP_{cooling}$ is the ratio of cooling capacity to work input.

6.1. Performance characteristics of basic systems

The most basic transcritical CO_2 heat pump (TCHP) system is comprised of an evaporator, compressor, gas cooler and expansion device only. The addition of certain components such as an accumulator or a suction line heat exchanger is not considered a major system modification, and in this review these systems will still be considered basic.

A large number of studies have modeled the operation of TCHPs to obtain a better understanding of the impacts of various operating parameters. Most models have been validated against experimental data or results of other published works.

In the supercritical region, the temperature and pressure are independent properties, thus the outlet temperature of the gas cooler is independent of the pressure. Several simulations revealed that an optimum gas cooler pressure exists for a given gas cooler outlet temperature [58–61]. Kauf [58] correlated the optimum pressure in terms of ambient temperature of air, which was the heat recovery fluid for the gas cooler. Liao et al. [59] derived a correlation for optimum pressure based on the evaporator temperature and the gas cooler outlet temperature. Similarly, Sarkar et al. [60] used gas cooler outlet temperature and evaporator temperature to develop correlations for optimum pressure, system COP and optimal gas cooler inlet temperature. Zhang et al. [61] analyzed the optimum discharge pressure using a system with two stage expansion via throttling valves.

The heat transfer process in the gas cooler is also very different from the condensing process of conventional refrigerants as shown in Fig. 2. Because heat transfer occurs by sensible cooling, the difference between the CO_2 temperature at the gas cooler inlet and outlet is typically larger than during heat rejection by condensation. This temperature difference is known as the refrigerant temperature glide. Compared to a condensation process, the gliding temperature profile of CO_2 can be more closely matched to the

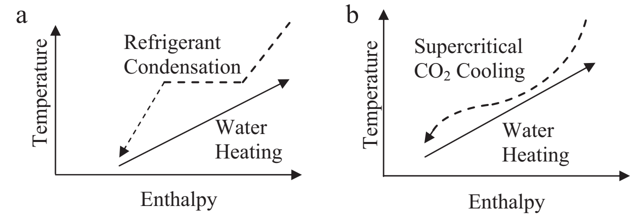

Fig. 2. Temperature profile for heat rejection by: (a) condensation process and (b) supercritical gas cooling process.

gliding temperature profile of the secondary fluid, which improves heat exchanger effectiveness.

The performance of a TCHP improves when the temperature glide increases. Results from a simulation by Laipradit et al. [62] showed that COP increased as water inlet temperature decreased. The reduced water inlet temperature corresponds with a reduced refrigerant temperature at the gas cooler outlet and hence a larger refrigerant temperature glide.

Confirming the impact of temperature glide, a theoretical and experimental study by Cecchinato et al. [63] showed that the COP of a TCHP increased significantly as the water temperature glide increased. An air-source CO_2 heat pump water heater was tested by heating water from 15 to 45 °C and 40 to 45 °C in separate trials. The trials were also run for three different gas cooler lengths. COP was 56–84% greater for the 15–45 °C temperature rise. The best performance improvements resulted from the gas cooler with the largest heat transfer area.

An experimental study by Fernandez et al. [64] examined the overall performance of a CO_2 system connected to a storage tank under three different heating conditions. The three scenarios represent: initial heating of a tank from 15 to 57.2 °C; reheating to 57.2 °C after cooling due to standby loss, and reheating to 57.2 °C after cold make-up water is added to the tank. The first scenario has the largest water temperature glide since the entire tank must be heated from15 to 57.2 °C. In scenario two the tank is at an average temperature of 42.2 °C with a small vertical temperature gradient in the tank. This represents the smallest temperature glide. In the third scenario the average tank temperature is 42.2 °C but significant stratification occurs in the tank due to the cold make-up water remaining in the bottom of the tank. Water in the bottom of the tank is at a temperature much lower than the average, indicating a medium temperature glide. Overall performance was best for initial heating and worst for standby loss reheating. Compared to initial tank heating, COPs were 30–40% lower for standby loss reheating.

Because CO_2 can provide a large temperature glide TCHP performance can actually benefit from multiple heating loads. Stene [65] performed an experimental and theoretical study on a dual function heat pump water heater. The schematic of the system is shown in Fig. 3. The gas cooler was partitioned to separately serve the functions of domestic water preheating, space heating and domestic water reheating. In this way, the temperature profile of the water through the gas cooler most closely matches that of CO_2 and thus the temperature glide can be used advantageously. COP was greatest for combined mode operation, slightly lower for domestic water heating only, and lowest for space heating only operation. This trend is opposite to that of a heat pump using a conventional refrigerant in which COP is best for space heating only and worst for dual mode. The authors of the study recommended that to achieve comparable performance to a heat pump using a hydro fluoro carbon refrigerant, the heating demand for hot water production must constitute at least 25% of the total annual heating demand of the residence. Also, the return temperature in the space heating system must be 30 °C or lower. Finally, the city water

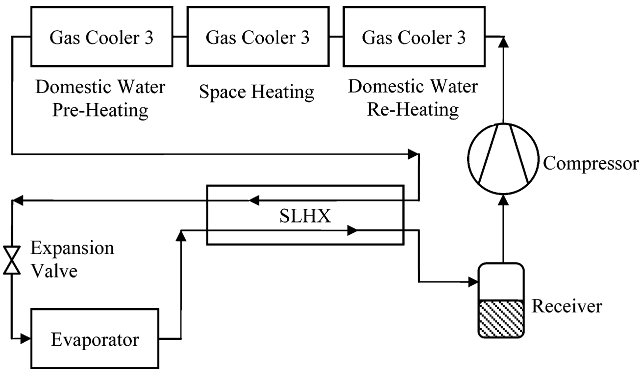

Fig. 3. Schematic diagram of TCHP with partitioned gas cooler for dual-function heating [65].

temperature must be 10 °C or lower and temperature stratification must be maintained in the domestic hot water storage tank.

A common addition to the most basic system is the suction line heat exchanger (SLHX). An SLHX brings the discharge vapor of the gas cooler into thermal exchange with the discharge vapor of the evaporator by means of a counter-flow heat exchanger, as shown in Fig. 3. The SLHX serves to subcool the gas cooler vapor before it goes through the expansion process. At the same time the evaporator vapor is superheated before it enters the compressor. A simulation performed by Robinson and Groll [66] obtained a 7% increase in COP when a SLHX was added to a basic transcritical CO_2 heat pump system.

The effects of an SLHX on the performance of a TCHP water heater were also investigated by Kim et al. [67]. The simulation model was validated against experimental data and the model was used to optimize SLHX size with respect to gas cooler pressure. The model revealed that system COP improves slightly with increased SLHX length, but only up to a certain gas cooler pressure. COP improved up to 4%. In addition, the authors concluded that the optimum gas cooler pressure decreases as SLHX length increases. These results confirm the findings of Chen and Gu [68] who determined that as SLHX effectiveness increases, the optimum pressure decreases and the COP increases.

White et al. [69] used a theoretical model to compare the performance of a system with an SLHX versus a system in which the heat exchange area of the gas cooler was increased but the SLHX was absent. Specifically, the heat transfer area of the gas cooler was increased by an amount equal to the heat transfer area of the SLHX; this increased the gas cooler size by 17%. The modification increased the optimum gas cooler pressure from 110 to 124 bar, which in turn affected a 20% increase in the heating capacity, though the COP remained unchanged. The mass flow rate also increased which is due to the higher density (due to lower temperature) of the vapor at the compressor inlet. The authors attribute the capacity increase primarily to the increased CO_2 flow rate.

A transcritical CO_2 heat pump can also be used to simultaneously provide a supply of heated water and cold water, thus maximizing efficiency. Sarkar et al. [70] theoretically analyzed a system designed for simultaneous heating and cooling of water. One stream of water served as the heat source and was thus cooled by the evaporator. A second stream of water served as the heat sink and was thus heated in the gas cooler. The two water streams entered the respective heat exchangers at the same temperature. The study analyzed the ratio between gas cooler area and evaporator area. For the test conditions the optimum ratio ranged from 1.6 to 1.9.

Another simulation involving simultaneous heating and cooling was performed by Byrne et al. [71]. The study compared the energy consumption of a simultaneous heating and cooling system to that of two individual heat pumps performing the heating and cooling operations separately. The simultaneous TCHP system consumed 27% less electricity than the two separate TCHPs operating under the same conditions.

The use of CO_2 for high temperature water heating was considered in an experimental and theoretical study by White et al. [69]. The experimental system was tested for water output temperatures of 65, 77.5 and 90 °C; the simulation model analyzed output temperatures as high as 120 °C. At evaporation temperatures of −6.4 °C, the maximum COP of the experimental system was approximately 3.0 for the 90 °C temperature. The model predicted a 21% decrease in COP and a 33% decrease in heating capacity as water temperature increased from 65 to 120 °C. Sarkar et al. [72] also investigated high temperature water heating and concluded that CO_2 was not recommended for output temperatures greater than 200 °C due to excessively high pressures. At an output temperature of 200 °C, the optimum discharge pressure was over 20 MPa.

One of the early experimental studies on TCHP water heaters was performed by Neksa et al. [73]. The prototype system included a suction line heat exchanger. The evaporator pressure and temperature were 35 bar and 0 °C respectively, and gas cooler pressure was 90 bar. The system operated with a COP of 4.3 while heating water from 8 °C to 60 °C. The COP decreased to 3.0 when the evaporator temperature was reduced to −20 °C.

Anstett [74] reported the performance of an air-source TCHP water heater installed in a hospital. The heat pump was designed to deliver hot water at temperatures between 60 and 80 °C with water inlet temperatures as low as 10 °C and ambient temperatures from −20 to 40 °C. COP ranged from 2 to 5 for the system. Performance was good even at low ambient temperatures, for instance, at an ambient temperature of −5 °C the system delivered water at 70 °C with a COP of 2.5.

Another application of transcritical CO_2 heat pumps is hydronic floor heating. Hihara [75] tested an experimental TCHP designed for domestic water heating and on-demand hydronic floor heating. The water heating took place at night when electricity cost is low. Hot water was stored in a large insulated storage tank. Performance was tested for various demand conditions and seasonal performance was calculated based on assumed daily demand models. The seasonal COP for the combined heating loads was 2.7. This value included motor efficiency and transient thermal losses from the tank.

The refrigerant charge volume is an important parameter for TCHP systems. Cho et al. [76] investigated the impacts of charge volume using an experimental system designed for cooling. $COP_{cooling}$ increased as charge volume increased to an optimum point. Beyond the optimum point, there is a slow reduction in $COP_{cooling}$ as the system becomes overcharged. Compared to other refrigerants, CO_2 performance was much more sensitive to under-charged conditions.

Several studies compared the performance of experimental systems using CO_2 versus analogous systems using other refrigerants [63,72,77–79]. Tamura et al. [77] built and tested a prototype automotive air conditioning system for heating, cooling and dehumidification using a transcritical CO_2 cycle. The design goal was to match or exceed both the heating and cooling performance of an analogous system using R134a. For tests under cooling conditions, $COP_{cooling}$ values were equal for CO_2 and R134a. For heating mode, the transcritical CO_2 system achieved a heating COP 1.31 times greater than the R134a system. Cecchinato et al. [63] also found CO_2's performance to be similar to that of R134a, but CO_2 actually performed better than R134a in applications with larger secondary-fluid temperature increase. Furthermore, results showed that in trials with large temperature increase, CO_2 benefited more than R134a from an increase in gas cooler (condenser) size.

The performance of a system using R410A was compared to that of a transcritical CO_2 system by Richter et al. [78]. The air-to-air systems were tested for different ambient conditions. COP values for the CO_2 system were generally lower than the R410A system; however, heating capacity was greater for CO_2 under most operating temperatures. One significant finding of the study was that as ambient temperature decreased, the heating capacity of the TCHP decreased marginally compared to the R410A system. Thus CO_2 systems would require less supplemental heat at lower ambient temperatures.

6.2. Component and system modifications

The improvement of overall cycle performance in a TCHP generally requires consideration of the system as a whole. The interactions between the components do not always allow for isolated improvements to a single component. Still, each component or process in the cycle plays a role in dictating the overall performance of the system.

Analysis the exergetic efficiency (2nd law efficiency) of TCHP provides insight as to which components hold the greatest potential for overall improvements of cycle performance. A theoretical analysis of 2nd law efficiencies by Robinson and Groll [66] compared the component irreversibilities (exergy losses) in a basic TCHP. The study showed that the expansion valve suffers the most irreversibilities followed in order by the compressor, gas cooler and evaporator. Yang et al. [80] also found that the most exergy loss occurred in the expansion valve, but concluded that the next greatest contribution to exergy loss depended on operating conditions. In some cases the losses of the gas cooler were greater than that of the compressor. In other cases the trend was reversed.

In contrast to Robinson and Groll [66] and Yang et al. [80], Sarkar et al. [81] concluded that the compressor had greatest exergy loss, followed in order by the gas cooler, evaporator and finally the expansion valve. The model of Sarkar et al. [81] was designed for simultaneous heating and cooling and included the effects of heat transfer and fluid flow. These factors may have led to the differences from the other studies.

Each process in the cycle can be modified in a variety of ways in the attempt to improve overall performance. Some modifications make isolated alterations of only one component. Other modifications are more closely tied to the interaction of system components. These changes may require the alteration more than one component. Sections 6.2.1–6.2.3 will address the possible process modifications which may lead to cycle improvements.

6.2.1. Heat exchanger modifications

Improvements to the gas cooler and the evaporator can significantly impact the performance and application potential of TCHPs. Pettersen et al. [82] reviewed unique characteristics of heat exchangers used for CO_2 in evaporators and gas coolers. By increasing the contact area between the refrigerant and the heat exchanger surface, microchannel tubes can reduce the overall size of a heat exchanger for a given heating or cooling capacity. The automotive market in particular would benefit from the reduced size and weight of micro-channel heat exchangers. Microchannel tubes are also capable of withstanding high working pressures, making them compatible with CO_2 systems.

The design of a gas cooler is strongly dependant on the secondary fluid used and the flow characteristics. Fronk and Garimella [83] drew attention to the importance of the ratio between CO_2 and secondary fluid heat transfer coefficients. Generally $h_{air} < h_{CO_2}$ and $h_{water} > h_{CO_2}$. Therefore in a water coupled heat exchanger the overall heat transfer coefficient is more sensitive to the CO_2 heat transfer coefficient; h_{CO_2} is the primary factor which determines overall heat transfer coefficient. In an air coupled heat exchanger the overall heat transfer coefficient is more sensitive to the heat transfer coefficient of air; h_{air} dominates the overall heat transfer coefficient.

Gas cooler performance improves when the temperature difference between CO_2 and the heat recovery fluid (air or water) at any point in the gas cooler is reduced. Sarkar et al. [60,81] analyzed the irreversibilities of the gas cooler and concluded that approximately 90% of the heat exchanger losses were due to temperature differences between the refrigerant and the secondary fluid. Irreversibilities due to pressure drop in the gas cooler were negligible in comparison.

In a counter-flow gas cooler, the difference between the CO_2 inlet temperature and the secondary fluid outlet temperature is known as the hot-side approach temperature difference (ΔT_{ap}). Cold-side ΔT_{ap} is the difference between CO_2 outlet temperature and secondary fluid inlet temperature. Fronk and Garimella [83,84] showed that as the cold-side ΔT_{ap} is reduced, the optimum gas cooler pressure is reduced. This leads to a reduction in compressor work.

Losses due to temperature difference can be reduced by increasing heat transfer area, but there is a limit to the effectiveness of this approach. A theoretical optimization of tube-in-tube heat exchanger geometry was performed by Sarkar et al. [85] based on minimizing irreversibility. The study showed that for a given set of operating conditions, a heat exchanger has an optimum capacity for a given diameter and length of the refrigerant tube. Below the optimum diameter, increased pressure drop leads to greater viscous losses. Increasing the diameter decreases the pressure drop but increases the thermal dissipation due to reduced heat transfer coefficient. Thermal dissipation can be reduced by increasing tube length, but this in turn leads to increased pressure drop. Hence an optimum set of dimensions exist which will provide the greatest heat transfer capacity.

Shell and tube heat exchangers for use with CO_2 were theoretically analyzed by Hwang and Radermacher [86]. Three shell and tube heat exchanger designs were modeled for gas cooling and evaporation processes. For each heat exchanger the performance and the component mass was compared to conventional R22 heat exchangers. The CO_2 gas cooler could achieve an equivalent capacity to the R22 condenser while reducing the mass by about 50%.

Experimental and theoretical studies by Fronk and Garimella [83,84] investigated a water-coupled gas cooler with serpentine pattern microchannel tubes. Water flowed through plate-type heat exchangers with internal fins, while CO_2 passed through the microchannels in a cross-counter-flow arrangement. The configuration facilitated the required heat capacity in a more compact size.

Air-coupled CO_2 gas coolers were also investigated [87–89]. Hwang et al. [87] tested the performance of a fin and tube gas cooler under various operating conditions. The test parameters were air inlet temperature, air velocity, refrigerant flow rate and gas cooler pressure. A series of twelve parametric tests were conducted as a first step in establishing a performance database.

A microchannel gas cooler was modeled by Yin et al. [88]. In the model, CO_2 flowed through microchannel tube-banks, while air was maintained in cross-flow conditions. Each tube bank consisted of ten or more parallel microchannel tubes connected to a header at each end. As a baseline a single tube-bank was tested. Two configurations were tested: in the first test, additional tube-banks were added in the plane perpendicular to the air flow (thus increasing the frontal area of the heat exchanger); in the second test, tube-banks were aligned in the direction the air flow, one behind the other, as shown in Fig. 4. In the first test, the model showed an increase in heating capacity from one to three sets of tube-banks. More than three tub-banks produced marginal increase in heat capacity. In

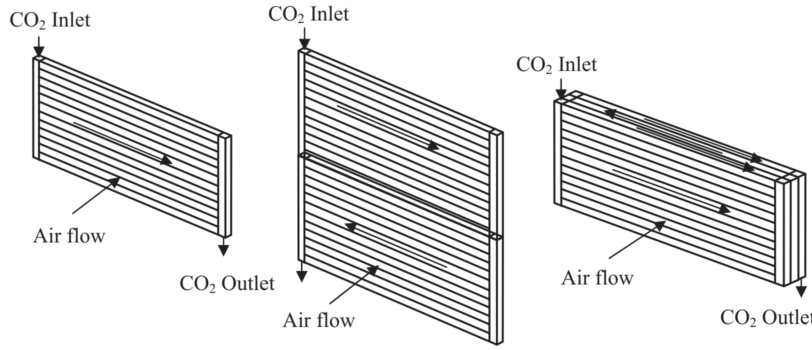

Fig. 4. Heat exchanger tube-bank configurations for microchannel gas cooler [88].

the second test, increasing the number of tube banks resulted in a decreased ΔT_{ap} and an increased heating capacity.

Neksa et al. [89] proposed a novel TCHP system in which the gas cooler rejects heat to air driven by natural convection instead of a fan. Both experimental and simulation studies were conducted. The counter flow gas cooler consisted of vertical aluminum fins for upward air flow bonded to vertical refrigerant tubes with high temperature CO_2 entering at the top. The experimental setup achieved cold-end ΔT_{ap} from 2.6 to 10.8 °C and a heat output between 71 and 140 W for different operating conditions. Further experimentation investigated the effects of an extended "chimney" attached to the top of the heat exchanger. This modification increased the air flow rate and the heat output. The simulation model was used to optimize fin spacing, thickness and height and to analyze other system modifications.

Research has also been conducted to investigate heat exchanger modifications for use in CO_2 evaporators. Bendaoud et al. [90] developed a model for the purpose of analyzing the performance of finned tube evaporators with CO_2. The model showed that the pressure drop of CO_2 through the evaporator is less than with other refrigerants. The use of a microchannel heat exchanger as the evaporator has also been shown to improve the performance of a TCHP system. Yun et al. [91] used a numerical model to compare a microchannel evaporator for air conditioning with a conventional fin-and-tube evaporator. The model was validated against the results of an experimental system using R134a and also validated against the results of a study by Beaver et al. [92]. The microchannel evaporator had a 33% greater heat transfer capacity. Compared to the conventional heat exchanger, the capacity of the microchannel heat exchanger was more strongly dependant on tube width, tube pitch and fin spacing.

Other authors have also developed models for the purpose of analyzing microchannel evaporators using CO_2. Kim and Bullard [93] developed a model to analyze the performance of a two-slab microchannel evaporator. The model incorporated existing correlations for heat transfer coefficient and pressure drop and can be used to aid in designing compact microchannel evaporators for use with CO_2. Jin et al. [94] developed their model using new correlations specifically developed for CO_2. The model was validated experimentally and was shown to be applicable for systems with high or even superheated vapor quality.

CO_2 is well suited for use with microchannel tubes because its high working pressure and high vapor density reduce the problem of phase maldistribution experienced by other refrigerants. Brix et al. performed a modeling study to analyze the effects of non-uniform CO_2 distribution [95]. CO_2 maldistribution is considered both due to non-uniform airflow and due to non-uniform phase distribution in the channels. For horizontal flow of refrigerant, uneven airflow distribution and non-uniform refrigerant quality at the inlet both caused reduction in the evaporator capacity. For vertical upward flow, airflow maldistribution caused capacity reduction, but refrigerant phase distribution did not impact the capacity. The authors note that for a system with more than two channels the results may prove different.

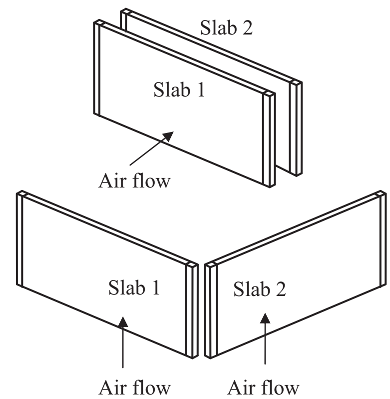

Fig. 5. Heat exchanger slab configurations for microchannel evaporator [91].

The alignment and orientation of the evaporator slabs also impacts the performance of a CO_2 evaporator. A simulation by Yun et al. [91] compared the performance of two cross-flow evaporator configurations using CO_2 as the refrigerant and air as the secondary fluid. Fig. 5 shows the two arrangements. Two slabs of microchannel tubes arranged in a V-shape showed better heat transfer capacity than two slabs arranged in series with respect to airflow.

An indirect modification of the evaporation process is the addition of a flash gas bypass (FGB) system. Fig. 6 shows the schematic diagram of a TCHP with FGB. In a FGB arrangement, the refrigerant

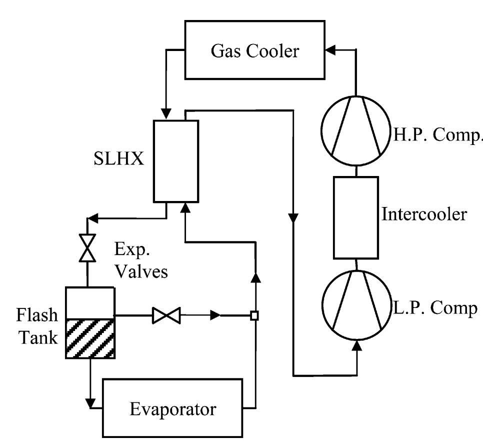

Fig. 6. Schematic diagram of TCHP with flash gas bypass [96].

expands through a single throttling device to the evaporator pressure and then enters a flash tank. From the flash tank only saturated liquid enters the evaporator while vapor bypasses the evaporator. The evaporation process is impacted because the vapor quality at the evaporator inlet is significantly reduced. An experimental study by Elbel and Hrnjak [96] showed that FGB increased the CO_2 evaporator's heat transfer capacity up to 9%, and $COP_{cooling}$ of the TCHP system improved up to 7%. The improved performance can mainly be attributed to reduced pressure drop through the evaporator and increased heat transfer coefficient in the evaporator.

In the same study, Elbel and Hrnjak [96] also investigated the impacts of a FGB arrangement on the CO_2 distribution in a microchannel evaporator. FGB led to more uniform refrigerant distribution in the evaporator.

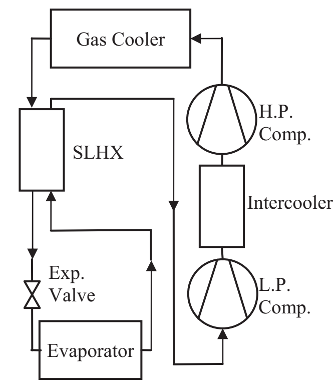

Fig. 7. TCHP with dual-compression and intercooling [97].

6.2.2. Compression process modifications

Efficiency of a TCHP can be improved by various modifications of the compression process. The most basic compressor modification is two-stage compression. By separating compression into two stages, each compressor will have a lower pressure ratio which in turn improves the isentropic efficiency. Further improvements in cycle efficiency can be obtained by cooling the refrigerant between the compression stages. Cecchinato et al. [97] showed that the use of intercooling in a transcritical CO_2 system with two-stage compression led to improved cycle efficiency. There are various methods of intercooling, and multi-stage compression systems can be configured in different ways as will be discussed in below.

A theoretical study by Cecchinato et al. [97] analyzed the performance of transcritical CO_2 air conditioning system incorporating dual compression with intercooling (DCWI). Fig. 7 shows the system configuration. Compared to a system using single-stage compression, DCWI improved the $COP_{cooling}$ 9%. At lower evaporator temperatures, $COP_{cooling}$ gains were more substantial. The results also showed that the benefits of inter stage cooling are strongly enhanced by the use of a suction line heat exchanger.

A common method of intercooling is to reject heat to ambient air by means of a heat exchanger. Cecchinato et al. [97] pointed out that due to the higher temperatures involved in the transcritical cycle, CO_2 is better able to take advantage of intercooling by rejecting heat to the environment than conventional refrigerants. Cavallini et al. [7] tested the cooling performance of an experimental system incorporating DCWI at different intercooler temperatures. The $COP_{cooling}$ at optimum discharge pressures was 2.1 for the 20.5 and 21.5 °C intercooler temperatures and 1.8 for the 33.0 °C temperature. A simulation model was also developed in order to analyze three modifications to the two-stage compression system. The addition of a SLHX provided a 7.6% improvement in $COP_{cooling}$. Incorporating two-stage throttling improved the $COP_{cooling}$ by 21.1%. Including both of these modifications increased the $COP_{cooling}$ by 24.0% above that of the baseline system.

Beyond DCWI, various systems can be configured which integrate modifications of both the expansion and compression processes. Flash intercooling is an alternative means of cooling the refrigerant between compression stages in which the inter-stage CO_2 temperature is reduced by mixing with expansion vapor in a flash tank. Fig. 8 shows the system schematic and cycle diagram of a TCHP incorporating flash intercooling. Agrawal and Bhattacharyya [98] determined that, unlike other methods of intercooling, two-stage compression with flash intercooling decreased the COP compared to that of an analogous system with single stage compression. This is due to the fact that mass flow rate through the second stage compressor increases significantly. Though the specific work of compression in the second stage is reduced, actual compression work in the second stage increases. Intermediate pressure was found to have little impact on COP.

Another modification is the use of an economizer tank, which is similar to the flash gas bypass system discussed in Section 6.2.1. Economizer systems in various configurations have been studied by several researchers [97,99–101]. Cho et al. [99] obtained an increase in $COP_{cooling}$ up to 16.5% by the addition of an economizer to an experimental dual compression transcritical CO_2 system. Cecchinato et al. [97] theoretically analyzed a similar system which resulted in a $COP_{cooling}$ increase of 16.8–28.7% compared to a system using single compression. Sarkar and Agrawal [100] tested a

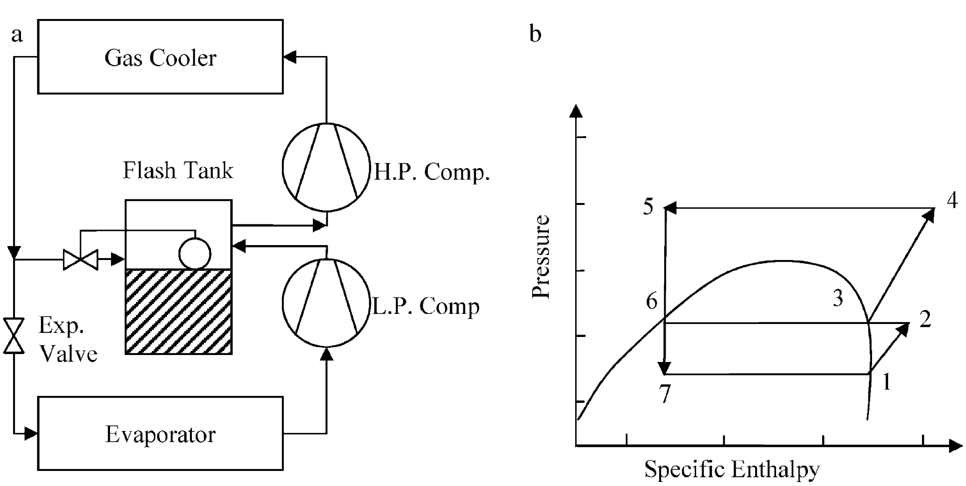

Fig. 8. TCHP with flash intercooling: (a) schematic diagram and (b) cycle *P–h* diagram [98].

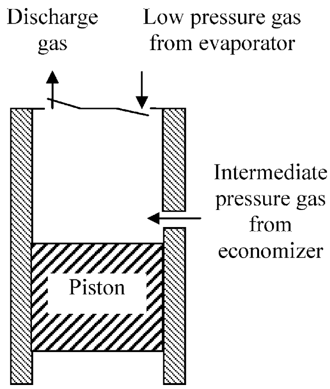

Fig. 9. Reciprocating piston compressor with Voorhees modification [101].

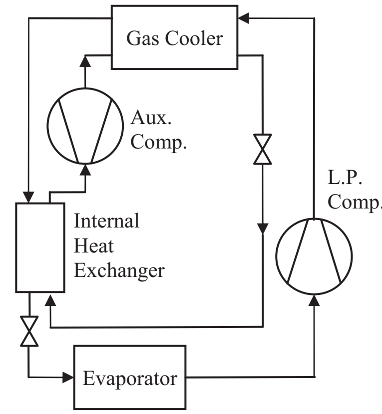

Fig. 11. Schematic diagram of an auxiliary loop TCHP system [97].

parallel compression system with an economizer. The $COP_{cooling}$ improved by 47.3% over a conventional transcritical system.

Zha et al. [101] tested an experimental laboratory setup which included a flash economizer and a single-stage reciprocating piston compressor with a Voorhees-type modification. In a Voorhees compressor, intermediate pressure CO_2 is injected into the compression chamber during only a portion of the intake stroke. The compressor modification is shown in Fig. 9. The Voorhees-economizer cycle was compared to a single stage compression cycle without economization. The economizer-cycle increased heating capacity for all operating conditions; however, COP improved only for low compressor speeds and low evaporator temperatures. At an evaporator temperature of −20 °C, the economizer provided nearly double the heating capacity.

A dual compression-dual expansion system with internal heat exchanger was experimentally tested for water heating by Fernandez et al. [64]. The performance was compared to a system that used only dual compression with intercooling. COP improved 7.5% for the test with a small increase in water temperature and an ambient temperature of 10 °C. At higher ambient temperatures and for larger temperature glide requirements, COP was virtually unchanged or actually decreased.

Cecchinato et al. [97] theoretically studied a system similar to Fernandez et al. [64] with the addition of a SLHX prior to the low pressure compressor, see Fig. 10. The other distinction is that the simulation was performed for cooling. The study concluded that $COP_{cooling}$ improved 15.6% compared to a DCWI system with an evaporator temperature of −30 °C. Furthermore $COP_{cooling}$ improved 20.6–29.3% for all conditions compared to a single compression system.

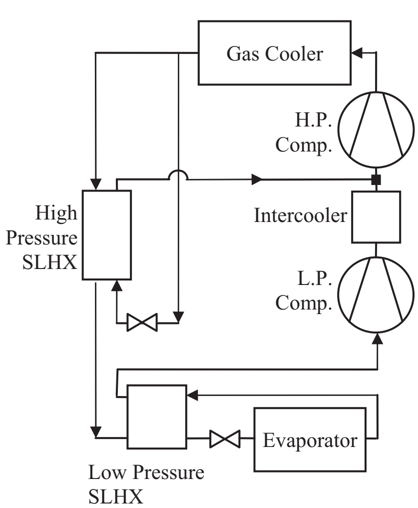

Fig. 10. Schematic diagram of a dual-compression dual-expansion system [97].

A transcritical system very unlike those of the previous studies was theoretically analyzed by Cecchinato et al. [97]. The system was composed of two separate heat pump loops with independent compressors. An internal heat exchanger served as the evaporator in the auxiliary loop while also subcooling the main-loop refrigerant prior to expansion. The system is represented in Fig. 11. Compared to heat pumps with more complex dual compression cycles, the auxiliary cycle's cooling performance was better only for low intermediate pressures.

6.2.3. Expansion process modifications

The two functions of the expansion device are to maintain the pressure difference between the gas cooler and the evaporator and to maintain proper flow of distribution to the evaporator. As discussed in Section 6.1, optimum system performance requires maintaining the optimum pressure in the gas cooler, therefore the expansion device plays an important role efficient system operation. An experimental study by Aprea and Maiorino [102] examined a method of maintaining the optimum high side pressure by two-stage expansion under changing ambient temperatures. The test setup incorporated a pressure valve and an electronically controlled expansion valve.

6.2.3.1. Capillary tube expansion. The primary means of expansion control for experimental transcritical CO_2 heat pump systems is through an expansion valve. Small refrigeration and heat pump units using conventional refrigerants commonly incorporate capillary tubes for expansion control, but there has been uncertainty about the effectiveness of capillary tubes in transcritical CO_2 systems. In order to ensure that a capillary tube can maintain the optimum gas cooler pressure and achieve optimal overall system performance theoretical studies have been conducted [103–109]. The studies generally indicate capillary tubes can perform adequately in TCHPs.

Agrawal and Bhattacharyya [103–105] performed a series of numerical studies on the flow characteristics of CO_2 in adiabatic capillary tubes and the performance of TCHP systems incorporating capillary expansion control. In one study [103] a capillary tube was used in a system designed for simultaneous heating and cooling. Performance was compared to a system studied by Sarkar et al. [70] which used a controllable expansion valve. Capillary length had significant influence on the gas cooler pressure and on the system COP. With optimal capillary length the system was tested for a range of gas cooler exit temperatures. The optimized capillary tube was able to very closely match the optimal gas cooler pressure calculated using the expansion valve. The COP for capillary

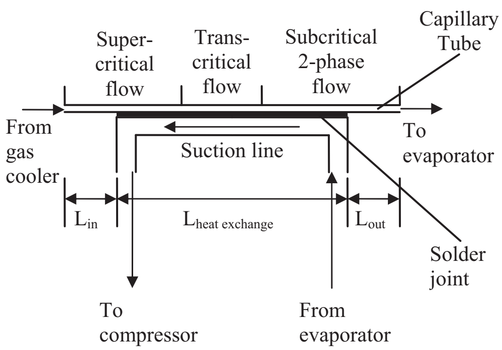

Fig. 12. Section of a non-adiabatic capillary tube heat exchanger [108].

and expansion valve systems were nearly the same and cooling capacity of the capillary system was slightly better. Optimal capillary tube length was found to be strongly dependant on capillary diameter and surface roughness.

Exergetic efficiency of a transcritical CO_2 system using a capillary tube for expansion control was also examined by Agrawal and Bhattacharyya [104]. The exergy efficiency of a TCHP with a capillary tube was 13–19% greater than that of a system using a controllable expansion valve operating with the same conditions. Exergy efficiency decreased under conditions of increased capillary diameter or conditions of increased capillary length.

Madsen et al. [106] also considered the performance of a capillary tube in a transcritical CO_2 cycle. A capillary tube was compared to a fixed pressure valve and an adjustable (optimized) expansion valve. Simulation was based on a constant enthalpy expansion model. Capillary performance was superior to the fixed pressure valve but the adjustable expansion valve produced the best COP. Authors concluded that for a small system, a capillary tube can be a competitive solution for expansion control.

In addition to adiabatic capillary tubes, non-adiabatic capillary tubes have been investigated as well [107–109]. In a non-adiabatic capillary tube, refrigerant will reject heat to the environment during the expansion in the tube. However, the heat rejected from a non-adiabatic capillary could be utilized as useful heat. As shown in Fig. 12, the capillary can serve as a heat exchanger by soldering a portion of the capillary to the refrigerant line between the evaporator outlet and the compressor inlet. This provides superheated vapor to the compressor in the same way as an SLHX. Chen and Gu [107] analyzed this type of system using used a non-adiabatic flow model to examine heat exchange and flow characteristics of a transcritical CO_2 capillary tube. Cooling capacity of the system was found to decrease as capillary length increased for a given inner diameter and pressure.

Non-adiabatic capillary tubes were also investigated by Agrawal and Bhattacharyya [108,109]. Flow characteristics, heat transfer characteristics and system performance were analyzed. Particular attention was paid to the effect of gas cooler outlet temperature, evaporator temperature and placement of the heat exchange portion of the tube relative to the flow condition of CO_2 in the tube (super critical, transcritical, or two-phase subcritical). It was recommended that the heat exchange take place through the two-phase region of the capillary.

6.2.3.2. Expansion work recovery. Using an expansion valve or capillary tube results in an energy loss because no useful work is done by the expansion process. These expansion losses associated with heat pumps using conventional refrigerants are generally small. In a transcritical CO_2 cycle, the greater pressure difference results in greater expansion losses, thus making work recovery more feasible and more beneficial. A theoretical investigation by Yang et al. [80] determined that the use of an expander in place of a conventional expansion valve produced a 50% decrease in exergy loss, resulting in a 30% improvement in system exergy efficiency. The expander reduced the optimum gas cooler pressure and led to a 33% higher $COP_{cooling}$.

Table 1
Theoretical $COP_{cooling}$ of transcritical CO_2 system and subcritical R22 system both incorporating expanders of various isentropic efficiencies [79].

Expander efficiency (%)	$COP_{cooling}$ at $T_{ambient}$ = 28 °C		$COP_{cooling}$ at $T_{ambient}$ = 50 °C	
	CO_2	R22	CO_2	R22
100	6.9–7.6	5.8	2.9–3.0	3.3
80	6.4–7.0	5.8	2.5–2.6	3.2
60	5.9–6.5	5.7	2.2–2.3	3.1
0	4.8–5.3	5.5	1.6	2.8

There are many devices which can potentially be used as expanders for the purpose of work recovery in a TCHP. In addition to turbines, expander types include rotary, reciprocating, scroll, screw, and vane. Each design has its advantages and limitations. Experimental and theoretical research has been conducted on a wide variety of systems.

Huff and Radermacher [79] carried out a theoretical study which compared the performance of reciprocating piston, rotary piston, and scroll expander/compressor pairs in a transcritical CO_2 cooling system. Each setup paired the same type of compressor and expander. The simulation modeled valve losses, internal leakage, internal heat transfer and unmatched volume ratio. Results indicated that the reciprocating and rotary piston expanders had similar performance, but the scroll expander performed poorly when leakage gaps were larger than 5 μm. The simulation also evaluated the performance of a CO_2 system with expander relative to a system using R22 with expander and with expansion valve. Table 1 compares the performance at ambient temperatures of 28 °C and 50 °C and expander efficiencies of 100%, 80%, 60%, and 0%. The CO_2 system generally performed better than the R22 system for lower ambient temperatures.

A numerical simulation of a CO_2 transcritical cooling cycle with a scroll type expander was performed by Kim et al. [110]. The system incorporated two-stage compression with intercooling using scroll compressors. The expander shaft was directly coupled to the first stage compressor. The model accounted for leakage by assuming a scroll clearance of 8 μm. At a gas cooler exit temperature of 35 °C and compressor inlet and outlet pressures of 35 bar and 100 bar respectively, the total efficiency of the expander was 54.4%. The expander efficiency was relatively stable for changing pressure ratios, however, the compressor efficiency decreased as inlet pressure increased. The expander reduced the required external compressor work by about 12%. This resulted in an increase in the cooling capacity of 8.6% and an increase in $COP_{cooling}$ of 23.5%.

Baek et al. [111] designed, built and tested an early prototype of a two-cylinder reciprocating piston work recovery expander for use with CO_2 under cooling conditions. Being a prototype, the system was not optimized and $COP_{cooling}$ values were actually less than one. Only one test condition included a supercritical pressure. For the transcritical test condition, the isentropic efficiency of the expander was 10.27%, and the expander improved the $COP_{cooling}$ 6.6% compared to the system using an expansion valve. A simulation model was also developed for the system [112]. The predicted exhaust pressure was significantly higher than measured in the experiments. This was attributed to a significant portion of CO_2 leaking from the cylinder via the piston ring in the experimental prototype.

A reciprocating piston expander was also tested by Nickl et al. [113]. The authors built and tested a three-stage piston expander which directly powered the second stage compressor in a CO_2

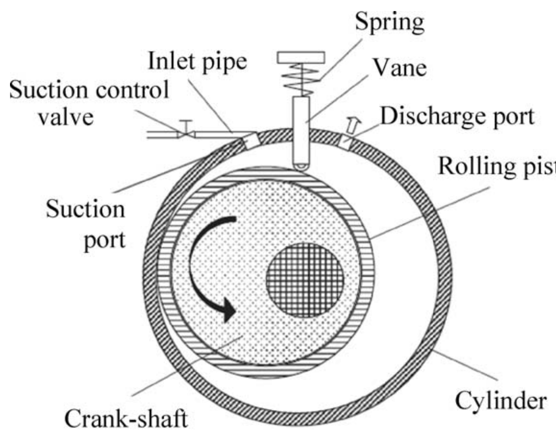

Fig. 13. Cross-section of a rolling piston expander [115].

refrigeration system. Isentropic efficiency of the expander ranged from 65 to 70%. The $COP_{cooling}$ increased by 40% compared to a transcritical CO_2 cycle with throttling valve. The expander also incorporated a liquid–vapor separator between the second and third stage expanders. This serves to minimize the losses which occur during the final expansion.

A reciprocating expander such as that tested by Nickl et al. [113] is referred to as a free piston expander because the piston's motion is not converted to rotational motion. One of the technical challenges for free piston expanders is controlling the intake and outlet in an efficient and effective manner. Electronic control is quite complicated, pneumatic control poses timing difficulties, and the lack of rotational motion does not facilitate cam driven valves. Zhang et al. [114] tested a prototype free piston expander–compressor with a novel slider-system as the inlet/outlet control mechanism. Test results showed that the control mechanism operated effectively for a range of pressure differences and pressure ratios. From the measured data, expander efficiency was estimated to be 62%.

Hua et al. [115] theoretically and numerically analyzed the performance of a rolling piston expander in a transcritical CO_2 cycle. Fig. 13 shows a section diagram of the rolling piston expander. The experimental system delivered a maximum expander efficiency of 45% and a maximum work recovery of 14.5%. The simulation model determined maximum work recovery of the expander was about 14–23% of the compressor work. As gas cooler outlet temperature increased the work recovery ratio increased but COP decreased. The experiment also showed that there is an optimal rotational speed of the expander which will maximize the expander efficiency, work recovery ratio and system COP.

The expansion work recovery of a swing piston expander was experimentally tested by Haiqing et al. [116]. To test the isentropic efficiency and work recovery, the expander was connected to an electrical generator with variable damped load. Maximum isentropic expander efficiency ranged from 28 to 44% depending upon operating conditions. The expander efficiency increased as the generator load increased from zero to an optimal load. Expander efficiency also increased as expander inlet temperature (gas cooler outlet temperature) increased. However, as gas cooler outlet temperature increased system COP and heating capacity decreased. Hence some of the highest expander efficiencies corresponded with low system COP values.

Another potential expander design is the revolving vane expander. Subiantoro and Ooi [117] compared the performance of four revolving vane expanders designed for use in a transcritical CO_2 cycle. Due to low friction, revolving vane expanders have high mechanical efficiency.

Table 2
Results of simulation comparing methods of transferring energy recovered from an expansion turbine [118].

Cycle (T_{ev} = 5 °C, $T_{gc,o}$ = 40 °C)	COP	Optimum high pressure (bar)	Inter-stage pressure (bar)
Single compression, expansion valve	2.418	100.4	–
Single compression, expander	3.211	96.14	–
DCOP	3.496	96.14	82.62
DCHP	3.521	101.3	80.90
DCLP	3.163	96.20	47.97

To make effective use of an expander, efficient energy transfer must occur. Employing gear box or generator systems results in energy loss. Alternatively, the expander can be directly connected to one of the system's compressors. Huff and Radermacher [79] note that for maximum utilization of the work recovered by the expander, the compressor should be directly coupled to the expander. Direct connection results in an efficient energy transfer; however, the work recovered by the expander varies with expander speed, and direct connection may not result in the optimum work recovery at all compressor speeds. Huff and Radermacher [79] determined that the speed ratio must be designed such that the system will operate close to the optimum for normal operating conditions.

A theoretical comparison was made by Yang et al. [118] between direct and indirect coupling of the expander and compressors in a transcritical CO_2 cooling system with dual compression. The investigation compared three configurations: (i) expander directly driving the high pressure compressor (DCHP); (ii) expander directly driving the low pressure compressor (DCLP); (iii) expander indirectly driving the low pressure compressor with optimized intermediate pressure (DCOP). Schematic diagrams of the three methods of energy transfer are shown in Fig. 14. The systems were also compared to single-stage compression systems with an expansion valve and with an expander. The results of the simulation are presented in Table 2. The best performance was achieved by the DCHP system. The DCOP system performed slightly worse. The DCLP system performed worse than a system with single stage compression system and expander. Optimum inter stage pressure was predicted to be much greater than the geometric mean pressure, which is typically used as the optimum intermediate pressure in a subcritical two stage compressor.

A suction line heat exchanger can be incorporated into a TCHP with expansion work recovery, but research has indicated that an SLHX is not effective at improving the system performance when a work recovery expander is used [66,79]. Robinson and Groll [66] used a simulation model to investigate the impacts of an SLHX on a system with an expander. A TCHP system with an expansion valve and SLHX was compared to a system with a work recovery expansion turbine. The expander system was then modified to include an SLHX. The expansion turbine system without the SLHX had an average COP 25% higher than that of the expansion valve/SLHX system. The use of both an SLHX and an expansion turbine actually reduced the COP up to 8%. Huff and Radermacher [79] also determined that a TCHP with a work recovery expander does not benefit from a SLHX. The heat exchanger becomes effective at increasing COP only if less than 30% of the work recovered by the expander can be used to reduce compressor work.

6.2.3.3. Ejector expansion. An alternative method to reduce expansion losses is the use of an ejector. An ejector reduces expansion losses and also increases the pressure at the compressor inlet, thus

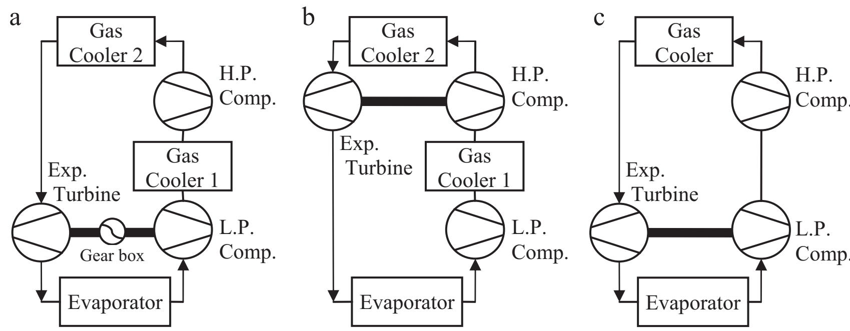

Fig. 14. Three methods of transferring work recovered from an expansion turbine: (a) indirect low pressure drive with optimized intermediate pressure; (b) direct high pressure drive; (c) direct low pressure drive [118].

reducing compressor work. Fig. 15 shows the schematic diagram and corresponding *P*–*h* diagram of a typical ejector cycle. The basic principles of ejector cycle are as follows. High pressure CO_2 from the gas cooler enters the nozzle of the ejector where its velocity is increased and pressure is decreased. This decreased pressure draws CO_2 vapor from the evaporator into the ejectors mixing chamber where the pressure increases. A diffuser is utilized to increase CO_2 pressure while also lowering the velocity. CO_2 then enters a liquid–vapor separator from which vapor is drawn into the compressor and liquid re-enters the evaporator.

An ejector expansion device was first investigated by Kornhauser [119] as an alternative throttling valve for a vapor compression cycle using R12 as a refrigerant. Use of an ejector in a transcritical CO_2 cooling cycle was later theoretically analyzed by Li and Groll [120] and Deng et al. [121]. Li and Groll [120] modeled a modified ejector cycle which was designed to adjust the vapor quality at the evaporator inlet and aid steady state operation. Deng et al. [121] focused on the importance of entrainment ratio for optimized performance. Both studies found significant improvements in $COP_{cooling}$.

Sarkar [122] theoretically analyzed the use of an ejector in a TCHP for simultaneous heating and cooling. The study modeled two systems: a typical ejector cycle and the modified ejector cycle of Li and Groll [120]. The performance difference between the two cycles was negligible except at low evaporator temperatures where the modified cycle was slightly less efficient. The ejector-expansion cycles were also compared to a conventional system with throttling valve and a system with a work recovery turbine. The ejector-expansion cycles both lowered the optimum discharge pressure compared to the throttling valve cycle. Both ejector cycles had COP values superior to the conventional cycle, but the system with the work recovery turbine had the best COP.

A prototype transcritical ejector system for cooling was built and tested by Elbel and Hrnjak [123]. Results were compared to a conventional system with expansion valve, and the results were used to validate a numerical model. The ejector improved $COP_{cooling}$ by up to 7%, and ejector efficiency ranged from 3.5 to 14.5%. Cycle performance was maximized at an optimum gas cooler pressure, but the greatest ejector efficiency occurred at a pressure much lower than this optimum gas cooler pressure.

A suction line heat exchanger decreases the inlet temperature of the ejector which increases both ejector efficiency and system efficiency. The impacts of an SLHX on a transcritical CO_2 system with an ejector were experimentally tested by Nakagawa et al. [124]. When testing the system without the SLHX, the vapor quality at the exit of the separator was approximately 0.9 for most of the test conditions. Thus the ejector system was allowing significant amounts of liquid to enter the compressor, and $COP_{cooling}$ was actually lower than an analogous system tested with an expansion valve. With a gas cooler outlet temp of 42 °C, the 60 cm suction line heat exchanger improved the ejector system's $COP_{cooling}$ by 55–63%. At higher gas cooler outlet temperatures, the performance improvement was even more significant.

7. Future potential of transcritical CO_2 heat pump systems

CO_2 has proven its potential as an appropriate working fluid for transcritical heat pump systems in the research setting. As discussed in the previous sections, the performance of CO_2 systems can match or exceed the performance of conventional refrigerants in many applications. In addition, there has already been some commercialization of transcritical CO_2 heat pump systems, the most notable of which are the Japanese heat pump water heaters which uses CO_2 in a transcritical cycle. The systems are manufactured by

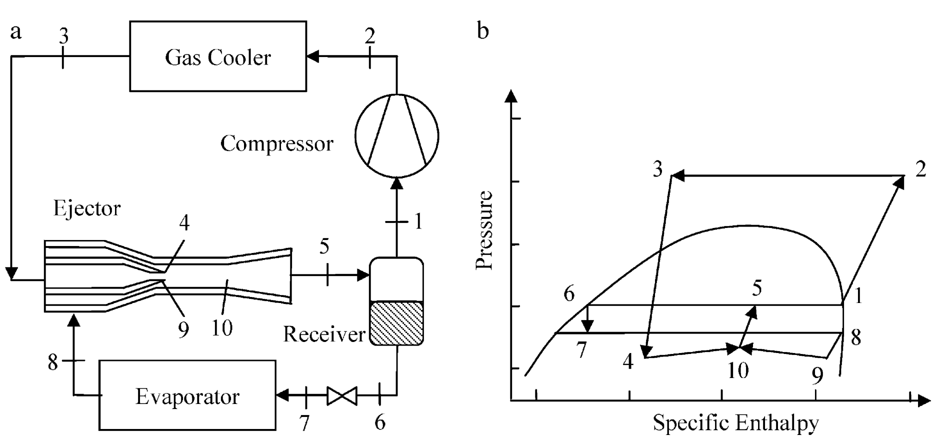

Fig. 15. TCHP with ejector expander: (a) system schematic and (b) cycle *P*–*h* diagram [122].

several companies and are marketed as the Eco Cute Heat Pump Water Heater. As of October 2009 more than 2 million units had been sold in Japan [15].

The Eco Cute system has shown that CO_2 heat pump systems have the potential for commercialization. It is likely that manufacturers will follow the success of the Eco Cute and will expand the use of CO_2 beyond residential water heaters.

One application which, to the authors' knowledge, has not been investigated in the academic or commercial setting is the use of CO_2 in direct expansion geothermal heat pumps. CO_2 holds the potential to provide both environmental and performance benefits in this application.

The majority of geothermal heat pump systems today use a dual loop system where a single phase brine solution circulates through the buried ground coils and a heat exchanger delivers the absorbed energy to the heat pump loop [125,126]. Less common are the direct expansion geothermal heat pumps (DXGHP) in which no secondary brine loop exists, but rather the refrigerant passes directly through the ground coils. Hence, evaporation takes place directly in the ground coils.

One reason a DXGHP is less common is the risk of ground water contamination by the refrigerant if a leak develops in the system [127]. Because it is a non-toxic natural substance, CO_2 does not pose any risk of ground water contamination. In addition, DXGHP systems require as much as 10 times more refrigerant, by volume, compared to brine based dual loop system [128]. This can significantly impact the cost of a DXGHP system using an expensive synthetic refrigerant. Because CO_2 is readily available and inexpensive, the increased charge volume would have less affect on cost.

Another issue with DXGHP systems is excessive pressure drop in the evaporator [125]. The long ground coils cause a large frictional pressure drop. As the evaporator's outlet pressure decreases the outlet temperature also decreases, thus reducing the system's heating capacity and COP. As will be explained below, a CO_2 system can tolerate a larger pressure drop without the significant decrease in evaporator temperature [82].

In the temperature range applicable for geothermal heat pumps, the change in temperature associated with a given change in saturation pressure (i.e. $\delta T/\delta P_{saturation}$) is much smaller for CO_2 than for other refrigerants. For example, at 0 °C a 1 kPa decrease in pressure reduces the saturation temperature of CO_2 by 0.01 °C while the same pressure drop causes the saturation temperature of R134a to decrease by 0.10 °C [6]. Thus assuming equal pressure drop through the ground coils of a DXGHP and equal inlet temperatures, CO_2 will have a greater outlet temperature than other refrigerants [82]. For this reason, CO_2 may be more suitable than other refrigerants for the large evaporative pressure drop associated with DXGHP systems.

Carbon dioxide has the potential to improve the performance, environmental impacts and economics of direct expansion geothermal heat pump systems. In order to gain a better understanding of the operating characteristics and feasibility of CO_2 in a DXGHP, further theoretical and experimental research is needed.

8. Conclusion

Carbon dioxide was used as a refrigerant in some of the earliest refrigeration systems. CO_2 fell out of favor when other refrigerants delivered superior performance over a wider range of conditions. In light of growing concerns over climate change, CO_2 has been revived as a potential refrigerant due to its benign environmental qualities. To overcome limitations imposed by CO_2's low critical point, most of the research has focused on the use of CO_2 in a transcritical cycle. Due to the high working pressures and the distinct nature of the transcritical cycle, optimization of system components and process parameters is ongoing.

Research has shown that transcritical CO_2 heat pumps can perform well for water and air heating applications. Furthermore, a significant body of research has shown that performance of the basic transcritical cycle can be improved by modifications such as dual-stage compression, multi-stage expansion, expansion work recover and heat exchanger optimization. As environmental regulations further restrict the use of harmful refrigerants, it is possible that transcritical CO_2 heat pumps will become much more common.

References

[1] U.S. Environmental Protection Agency. High GWP gases and climate change. U.S. Environmental Protection Agency, www.epa.gov/highgwp/scientific.html; 2010 [accessed 25.07.10].

[2] Intergovernmental Panel on Climate Change. Climate Change 2001: Working Group I: The Scientific Basis. Intergovernmental Panel on Climate Change; 2001, 6.12.2.

[3] Emissions from air conditioning systems in motor vehicles. http://europa.eu/legislation_summaries/internal_market/single_market_for_goods/motor_vehicles/interactions_industry_policies/l24280_en.htm; September 25, 2007 [accessed 05.10.10].

[4] Directive 2006/40/EC of the European Parliament and of the Council of 17 May 2006 relating to emissions from air-conditioning systems in motor vehicles and amending Council Directive 70/156/EEC. Journal of the European Union June 2006;161:12–8.

[5] Neksa P. CO_2 heat pump systems. International Journal of Refrigeration 2002;25:421–7.

[6] Kim MH, Pettersen J, Bullard CW. Fundamental process and system design issues in CO_2 vapor compression systems. Progress in Energy and Combustion Science 2004;30.

[7] Cavallini A, Cecchinato L, Corradi M, Fornasieri E, Zilio C. Two-stage transcritical carbon dioxide cycle optimization: a theoretical and experimental analysis. International Journal of Refrigeration 2005;28:1274–83.

[8] U.S. Secretary of Commerce. Thermophysical properties of fluid systems, http://webbook.nist.gov/chemistry/fluid; 2008 [accessed 14.05.10].

[9] Pearson A. Carbon dioxide—new uses for an old refrigerant. International Journal of Refrigeration 2005;28:1140–8.

[10] Calm JM. The next generation of refrigerants—historical review, considerations, and outlook. International Journal of Refrigeration 2008;31:1123–33.

[11] Lorentzen, G. Trans-critical vapour compression cycle device. Patent WO/07683; 1990.

[12] Lorentzen G, Pettersen J. A new, efficient and environmentally benign system for car air-conditioning. International Journal of Refrigeration 1993;16:4–12.

[13] Pettersen J. An efficient new automobile air conditioning system based on carbon dioxide vapor compression. ASHRAE Transactions 1994;100:657–65.

[14] EcoCute tops one million units. IEA Heat Pump Centre Newsletter 2007:13.

[15] EcoCute shipments exceed 2 million landmark. R744.com, http://www.r744.com/articles/2009-11-26-eco-cute-shipments-exceed-2-million-landmark.php; November 26, 2009 [accessed 04.08.10].

[16] BeyondHFCs. BeyondHFCs position paper on CO_2 vending machines. BeyondHFCs, http://www.beyondhfcs.org/search.result.php?Keyword=BeyondHFCs+position+paper+on+CO2+vending+machines; March 20, 2010 [accessed 24.10.10].

[17] Hepbasli A, Kalinci Y. A review of heat pump water heating systems. Renewable and Sustainable Energy Reviews 2009;13:1211–29.

[18] American Society of Mechanical Engineers. The Equitable Building Heat Pump System Dedication Ceremony Program. American Society of Mechanical Engineers, http://files.asme.org/ASMEORG/Communities/History/Landmarks/5541.pdf; May 8, 1980 [accessed 04.08.10].

[19] Hua T, Zhao Y, MinXia L, YiTai MA. Research and application of CO_2 refrigeration and heat pump cycle. Science in China Series E: Technological Sciences 2009;52(June (6)):1563–75.

[20] Staub J, Rasmusen BD, Robinson M. CO_2 as refrigerant: the transcritical cycle. ACHR News 2004;January 26.

[21] Asinari P, Cecchinato L, Fornasieri E. Effects of thermal conduction in microchannel gas coolers for carbon dioxide. International Journal of Refrigeration 2004;27:577–86.

[22] Oritz TM, Li D, Groll EA. Evaluation of the performance potential of CO_2 as a refrigerant in air-to-air air conditioners and heat pumps: system modeling and analysis. Final Report. ARTI; 2003.

[23] Yang J, Ma Y, Liu S, Zeng X. Comparison investigation on the heat transfer characteristics for supercritical CO_2 fluid and conventional refrigerants. In: 7th IIR Gustav Lorentzen conference on natural working fluids. 2006.

[24] Krasnoshchekov EA, Kuraeva IV, Protopopov VS. Local heat transfer of carbon dioxide at supercritical pressure under cooling conditions (translated from Teplofizika Vysokikh Temperatur) 1970;7: 856–62.

[25] Baskov VL, Kuraeva IV, Protopopov VS. Heat transfer with turbulent flow of a liquid at supercritical pressures in tubes under cooling conditions (translated from Teplofizika Vysokikh Temperatur) 1977;15: 81–6.

[26] Petrov NE, Popov VN. Heat transfer and resistance of carbon dioxide being cooled in the supercritical region. Thermal Engineering 1985;32: 131–4.
[27] Petukhov BS, Popov VN. Theoretical calculation of heat exchange and frictional resistance in turbulent flow in tubes of an incompressible fluid with thermophysical properties. High Temperature 1963;I:69–83.
[28] Yoon SH, Kim JH, Hwang YW, Kim MS, Min K, Kim Y. Heat transfer and pressure drop characteristics during the in-tube cooling process of carbon dioxide in the supercritical region. International Journal of Refrigeration 2003;26:857–64.
[29] Dang C, Hihara E. In-tube cooling heat transfer of supercritical carbon dioxide, part 1. Experimental measurement. International Journal of Refrigeration 2004;27:736–47.
[30] Pettersen J, Rieberer R, Leister A. Heat transfer and pressure drop characteristics of supercritical carbon dioxide in microchannel tubes under cooling. In: Preliminary proceedings of the 4th IIR-Gustav Lorentzen conference on natural working fluids. 2000. p. 99–106.
[31] Kuang G, Ohadi MM, Zhao Y. Experimental study on gas cooling heat transfer for supercritical CO_2 in microchannels. In: Second international conference on microchannels and minichannels. 2004. p. 325–32.
[32] Huai XL, Koyama S, Zhao TS. An experimental study of flow and heat transfer of supercritical carbon dioxide in multi-port mini channels under cooling conditions. Chemical Engineering Science 2005;60:3337–45.
[33] Huai XL, Koyama S. Heat transfer characteristics of supercritical CO_2 flow in small-channeled structure. Experimental Heat Transfer 2007;20:19–33.
[34] Liao SM, Zhao TS. Measurement of heat transfer coefficients from supercritical carbon dioxide flowing in horizontal mini/micro channels. Journal Heat Transfer 2002;124:413–20.
[35] Pitla SS, Groll EA, Ramadhyani S. New correlation to predict the heat transfer coefficient during in-tube cooling of turbulent supercritical CO_2. International Journal of Refrigeration 2002;25:887–95.
[36] Olson DA. Heat transfer of supercritical carbon dioxide flowing in a cooled horizontal tube. In: Proceedings of 4th IIR-Gustav Lorentzen conference on natural working fluids. 2000. p. 251–8.
[37] Kuang G, Ohadi M. Semi-empirical correlation of gas cooling heat transfer of supercritical carbon dioxide in microchannels. HVAC&R Research 2008;14:861–70.
[38] Kim JH, Yoon MS, Kim Y. An experimental investigation of heat transfer characteristics during in-tube gas cooling process of carbon dioxide. (IIF-IIR Commission B1) B6 2001: 17–21.
[39] Oh HK, Son CH. New correlation to predict the heat transfer coefficient in-tube cooling of supercritical CO_2 in horizontal macro-tubes. Experimental Thermal and Fluid Science 2010;34:1230–41.
[40] Petrov NE, Popov VN. Heat transfer and resistance of carbon dioxide being cooled in the supercritical region. Thermal Engineering 1985;32:131–4.
[41] Pettersen J, Rieberer R, Munkejord ST. Heat transfer and pressure drop for flow of supercritical and subcritical CO_2 in microchannel tubes. Final Technical Report. European Research Office of the U.S. Army, United States Army, London, England; 2000.
[42] Son CH, Park SJ. An experimental study on heat transfer and pressure drop characteristics of carbon dioxide during gas cooling process in a horizontal tube. International Journal of Refrigeration 2006;29:539–46.
[43] Cheng L, Ribatski G, Thome JR. Analysis of supercritical CO_2 cooling in macro- and micro-channels. International Journal of Refrigeration 2008;31:1301–16.
[44] Yoon SH, Cho ES, Hwang YW, Kim MS, Min K, Kim Y. Characteristics of evaporative heat transfer and pressure drop of carbon dioxide and correlation development. International Journal of refrigeration 2004;27:111–9.
[45] Bredesen AM, Hafner AH, Pettersen J, Neksa P, Aflekt K. Heat transfer and pressure drop for in-tube evaporation of CO_2. College Park: IIF-IIR-Commission B1, with E1 & E2; 1997.
[46] Cho JM, Kim MS. Experimental studies on the evaporative heat transfer and pressure drop of CO_2 in smooth and micro-fin tubes of the diameters of 5.00 and 9.52 mm. International Journal of Refrigeration 2007;30:986–94.
[47] Yun R, Kim Y, Kim MS, Choi Y. Boiling heat transfer and dryout phenomenon of CO_2 in horizontal smooth tube. International Journal of Heat and Mass Transfer 2003;46:2353–61.
[48] Zhao X, Bansal PK. Flow boiling heat transfer characteristics of CO_2 at low temperatures. International Journal of Refrigeration 2007;30:937–45.
[49] Oh HK, Ku HG, Roh GS, Park SJ, Son CH. Flow boiling heat transfer characteristics of carbon dioxide in a horizontal tube. Applied Thermal Engineering 2008;28:1022–30.
[50] Park CY, Hrnjak PS. CO_2 and R410A flow boiling heat transfer, pressure drop, and flow pattern at low temperatures in a horizontal smooth tube. International Journal of Refrigeration 2007;30:166–78.
[51] Wu XM, Zhao HY, Wang WC, Jing L, Zhang L. Experimental study on evaporating heat transfer of CO_2 in thin tube. Journal of Engineering Thermophysics 2005;26:823–5.
[52] Schael AE, Kind M. Flow pattern and heat transfer characteristics during flow boiling of CO_2 in a horizontal micro-fin tube and comparison with smooth tube data. International Journal of Refrigeration 2005;28:1186–95.
[53] Mastrullo R, Mauro AW, Rosato A, Vanoli GP. Carbon dioxide heat transfer coefficients and pressure drops during flow boiling: assessment of predictive methods. International Journal of Refrigeration 2010;33:1068–85.
[54] Cheng L, Ribatski G, Moreno Quiben Q, Thome JR. New prediction methods for CO_2 evaporation inside tubes: part I—a two-phase flow pattern map and a flow pattern based phenomenological model for two-phase flow frictional pressure drops. International Journal of Heat and Mass Transfer 2008;51: 111–24.
[55] Jung DS, Radermacher R, McLinden M, Didon D. A study of flow boiling heat transfer with refrigerant mixtures. International Journal of Heat and Mass Transfer 1989;32:1751–64.
[56] Muller-Steinhagen H, Heck K. A simple friction pressure correlation for two-phase flow in pipes. Chemical Engineering Process 1986;20: 297–308.
[57] Friedel L. Improved friction drop correlation for horizontal and vertical two-phase pipe flow. In: European two-phase flow group meeting. 1979. Paper E2.
[58] Kauf F. Determination of the optimum high pressure for transcritical CO_2-refrigeration cycles. International Journal of Thermal Sciences 1999;38:325–30.
[59] Liao SM, Zhao TS, Jakobsen A. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles. Applied Thermal Engineering 2000;20:831–41.
[60] Sarkar J, Bhattacharyya S, Gopal MR. Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications. International Journal of Refrigeration 2004;27:830–8.
[61] Zhang XP, Fan XW, Wang FK, Shen HG. Theoretical and experimental studies on optimum heat rejection pressure for a CO_2 heat pump system. Applied Thermal Engineering 2010.
[62] Laipradit P, Tiansuwan J, Kiatsiriroat T, Aye L. Theoretical performance analysis of heat pump water heaters using carbon dioxide as refrigerant. International Journal of Energy Research 2007;32(4):356–66.
[63] Cecchinato L, Corradi M, Fornasieri E, Zamboni L. Carbon dioxide as refrigerant for tap water heat pumps: a comparison with the traditional solution. International Journal of Refrigeration 2005;28:1250–8.
[64] Fernandez N, Hwang Y, Radermacher R. Comparison of CO_2 heat pump water heater performance with baseline cycle and two high COP cycles. International Journal of Refrigeration 2010;33:635–44.
[65] Stene J. Residential CO_2 heat pump system for combined space heating and hot water heating. International Journal of Refrigeration 2005;28:1259–65.
[66] Robinson DM, Groll EA. Efficiencies of transcritical CO_2 cycles with and without an expansion turbine. International Journal of Refrigeration 1998;21(7):577–89.
[67] Kim SG, Kim YJ, Lee G, Kim MS. The performance of a transcritical CO_2 cycle with an internal heat exchanger for hot water heating. International Journal of Refrigeration 2005;28:1064–72.
[68] Chen Y, Gu J. The optimum high pressure for CO_2 transcritical refrigeration systems with internal heat exchangers. International Journal of Refrigeration 2005;28:1238–49.
[69] White SD, Yarrall MG, Cleland DJ, Hedley RA. Modeling the performance of a transcritical CO_2 heat pump for high temperature heating. International Journal of Refrigeration 2002;25:479–86.
[70] Sarkar J, Bhattacharyya S, Gopal MR. Simulation of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications. International Journal of Refrigeration 2006;29:735–43.
[71] Byrne P, Miriel J, Lenat Y. Design and simulation of a heat pump for simultaneous heating and cooling using HFC or CO_2 as a working fluid. International Journal of Refrigeration 2009;32:1711–23.
[72] Sarkar J, Bhattacharyya S, Gopal MR. Natural refrigerant-based subcritical and transcritical cycles for high temperature heating. International Journal of Refrigeration 2007;30:3–10.
[73] Neksa P, Rekstad H, Zakeri GR, Schiefloe PA. CO_2 heat pump water heater: characteristics, system design and experimental results. International Journal of Refrigeration 1998;21:172–9.
[74] Anstett P. Measurement of the performance of an air/water heat pump using CO_2 or R744 for the production of hot water for use in a hospital. IEA Heat Pump Centre Newsletter 2006:35–8.
[75] Hihara E. Performance test of a carbon dioxide heat pump for combined domestic hot water and floor heating. IEA Heat Pump Centre Newsletter 2006:21–3.
[76] Cho H, Ryu C, Kim Y, Kim HY. Effects of refrigerant charge amount on the performance of a transcritical CO_2 heat pump. International Journal of Refrigeration 2005;28:1266–73.
[77] Tamura T, Yakumaru Y, Nishiwaki F. Experimental study on automotive cooling and heating air conditioning system using CO_2 as a refrigerant. International Journal of Refrigeration 2005;28:1302–7.
[78] Richter MR, Song SM, Yin JM, Kim MH, Bullard CW, Hrnjak PS. Experimental results of transcritical CO_2 heat pump for residential application. Energy 2003;28:1005–19.
[79] Huff HJ, Radermacher R. CO_2 compressor–expander analysis. Arlington, VA: Air-Conditioning and Refrigeration Technology Institute; 2003.
[80] Yang JL, Ma YT, Li MX, Guan HQ. Exergy analysis of transcritical carbon dioxide refrigeration cycle with an expander. Energy 2005;30:1162–75.
[81] Sarkar J, Bhattacharyya S, Gopal MR. Transcritical CO_2 heat pump systems: exergy analysis including heat transfer and fluid flow effects. Energy Conversion and Management 2005;46:2053–67.
[82] Pettersen J, Hafner A, Skaugen G. Development of compact heat exchangers for CO_2 air-conditioning systems. International Journal of Refrigeration 1998;21:180–93.
[83] Fronk BM, Garimella S. Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: part II—model development and validation. International Journal of Refrigeration 2011;34:17–28.

[84] Fronk BM, Garimella S. Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: part I—experiments. International Journal of Refrigeration 2011;34:7–16.
[85] Sarkar J, Bhattacharyya S, Gopal MR. Irreversibility minimization of heat exchangers for transcritical CO_2 systems. International Journal of Thermal Sciences 2009;48:146–53.
[86] Hwang Y, Radermacher R. Theoretical evaluation of carbon dioxide refrigeration cycle. HVAC&R Research 1998;4(3):245–63.
[87] Hwang Y, Radermacher R, Jin D, Hutchins JW. Performance measurement of CO_2 heat exchangers. ASHRAE Transactions 2005;111:306–16.
[88] Yin JM, Bullard CW, Hrnjak PS. R-744 gas cooler model development and validation. International Journal of Refrigeration 2001;24:692–701.
[89] Neksa P, Hoggen RL, Aflekt K, Jakobsen A, Skaaugen G. Fan-less heat exchanger concept for CO_2 heat pump systems. International Journal of Refrigeration 2005;28:1205–11.
[90] Bendaoud A, Ouzzane M, Aidoun Z, Galanis N. A new modeling approach for the study of finned coils with CO_2. International Journal of Thermal Sciences 2010;49:1702–11.
[91] Yun R, Kim Y, Park C. Numerical analysis on a microchannel evaporator designed for CO_2 air-conditioning systems. Applied Thermal Engineering 2007;27:1320–6.
[92] Beaver AC, Yun MJ, Bullard CW, Hrnjak PS. An experimental investigation of transcritical carbon dioxide systems for residential air conditioning. ARCR CR-18; 1999.
[93] Kim MH, Bullard CW. Development of a microchannel evaporator model for a CO_2 air-conditioning system. Energy 2001;26:931–48.
[94] Jin J, Chen J, Chen Z. Development and validation of a microchannel evaporator model for CO_2 air-conditioning system. Applied Thermal Engineering 2011;31:137–46.
[95] Brix W, Koern MR, Elmegaard B. Modeling distribution of evaporating CO_2 in parallel minichannels. International Journal of Refrigeration 2010;33:1086–94.
[96] Elbel S, Hrnjak P. Flash gas bypass for improving the performance of transcritical R744 systems that use microchannel evaporators. International Journal of Refrigeration 2004;27(27):724–35.
[97] Cecchinato L, Chiarello M, Corradi M, Fornasieri E, Minetto S, Stringari P, et al. Thermodynamic analysis of different two-stage transcritical carbon dioxide cycles. International Journal of Refrigeration 2009;32:1058–67.
[98] Agrawal N, Bhattacharyya S. Studies on a two stage transcritical carbon dioxide heat pump cycle with flash intercooling. Applied Thermal Engineering 2007;27:299–305.
[99] Cho H, Baek C, Park C, Kim Y. Performance evaluation of a two-stage CO_2 cycle with gas injection in the cooling mode operation. International Journal of Refrigeration 2009;32:40–6.
[100] Sarkar J, Agrawal N. Performance optimization of transcritical CO_2 cycle with parallel compression economization. International Journal of Thermal Sciences 2010;49:838–43.
[101] Zha S, Hafner A, Neksa P. Investigation of R-744 Voorhees transcritical heat pump system. International Journal of Refrigeration 2008;31:16–22.
[102] Aprea C, Maiorino A. Heat rejection pressure optimization for a carbon dioxide split system: an experimental study. Applied Energy 2009;86:2373–80.
[103] Agrawal N, Bhattacharyya S. Optimized transcritical CO_2 heat pumps: performance comparison of capillary tubes against expansion valves. International Journal of Refrigeration 2008;31:388–95.
[104] Agrawal N, Bhattacharyya S. Exergy assessment of an optimized capillary tube-based transcritical CO_2 heat pump system. International Journal of Energy Research 2009;33:1278–89.
[105] Agrawal N, Bhattacharyya S. Adiabatic capillary tube flow of carbon dioxide in a transcritical heat pump cycle. International Journal of Energy Research 2007;31:1016–30.
[106] Madsen KB, Poulsen CS, Wiesenfarth M. Study of capillary tubes in a transcritical CO_2 refrigeration system. International Journal of Refrigeration 2005;28:1212–8.
[107] Chen Y, Gu J. Non-adiabatic capillary tube flow of carbon dioxide in a novel refrigeration cycle. Applied Thermal Engineering 2005;25:1670–83.
[108] Agrawal N, Bhattacharyya S. Non-adiabatic capillary tube flow of carbon dioxide in a transcritical heat pump cycle. Energy Conversion and Management 2007;48:2491–501.
[109] Agrawal N, Bhattacharyya S. Performance evaluation of a non-adiabatic capillary tube in a transcritical CO_2 heat pump cycle. International Journal of Thermal Sciences 2008;47:423–30.
[110] Kim HJ, Ahn JM, Cho SO, Cho KR. Numerical simulation on scroll expander–compressor unit for CO_2 trans-critical cycles. Applied Thermal Engineering 2008;28:1654–61.
[111] Baek JS, Groll EA, Lawless PB. Piston-cylinder work producing expansion device in a transcritical carbon dioxide cycle. Part I: experimental investigation. International Journal of Refrigeration 2005;28:141–51.
[112] Baek JS, Groll EA, Lawless PB. Piston-cylinder work producing expansion device in a transcritical carbon dioxide cycle. Part II: theoretical model. International Journal of Refrigeration 2005;28:152–64.
[113] Nickl J, Will G, Quack H, Kraus WE. Integration of a three-stage expander into a CO_2 refrigeration system. International Journal of Refrigeration 2005;28:1219–24.
[114] Zhang B, Peng X, He Z, Xing Z, Shu P. Development of a double acting free piston expander for power recovery in transcritical CO_2 cycle. Applied Thermal Engineering 2007;27:1629–36.
[115] Hua T, Yitai M, Minxia L, Wei W. Study on expansion power recovery in CO_2 transcritical cycle. Energy Conversion and Management 2010;51:2516–22.
[116] Haiqing G, Yitaai M, Minxia L. Some design features of a CO_2 swing piston expander. Applied Thermal Engineering 2006;26:237–43.
[117] Subiantoro A, Ooi KT. Design analysis of the novel revolving vane expander in a transcritical carbon dioxide refrigeration system. International Journal of Refrigeration 2010;33:675–85.
[118] Yang JL, Ma YT, Liu SC. Performance investigation of transcritical carbon dioxide two-stage compression cycle with expander. Energy 2007;32:237–45.
[119] Kornhauser AA. The use of an ejector as a refrigerant expander. In: Proceedings of the 1990 USNC/IIR-Purdue refrigeration conference. 1990. p. 10–9.
[120] Li D, Groll EA. Transcritical CO_2 refrigeration cycle with ejector-expansion device. International Journal of Refrigeration 2005;28:766–73.
[121] Deng J, Jiang P, Lu T, Lu W. Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector. Applied Thermal Engineering 2007;27:381–8.
[122] Sarkar J. Optimization of ejector-expansion transcritical CO_2 heat pump cycle. Energy 2008;33:1399–406.
[123] Elbel S, Hrnjak P. Experimental validation of a prototype ejector designed to reduce throttling losses encountered in transcritical R744 system operation. International Journal of Refrigeration 2008;31:411–22.
[124] Nakagawa M, Marasigan AR, Matsukawa T. Experimental analysis on the effect of internal heat exchanger in transcritical CO_2 refrigeration cycle with two-phase ejector. International Journal of Refrigeration; in press.
[125] Halozan H. Direct-expansion ground-coupled heat pumps. IEA Heat Pump Centre Newsletter 2005;23(4):30–2.
[126] Omer A. Ground-source heat pumps systems and applications. Renewable and Sustainable Energy Reviews 2008;12:344–71.
[127] Wang X, Ma C, Lu Y. An experimental study of a direct expansion ground-coupled heat pump system in heating mode. International Journal of Energy Research 2009;33:1367–83.
[128] Prairie Energy Solutions. Direct Expansion Heat Pumps, http://www.pesdistributors.com/prod_detail.asp?ID=7097; 2010 [accessed 01.09.10].